Airside Economizer Low Limit Effect on Energy and Thermal Comfort

By

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ABSTRACT

Samantha Bakke (M.S., Architectural Engineering)

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Thesis directed by Professor Brian A. Rock

A continuous effort exists within the heating, ventilating, and air conditioning industry to not only enhance thermal comfort within indoor environments, but also for developing more energy efficient systems. An airside economizer can assist with the latter. Guidance is lacking for these devices in regards to the optimal airside economizer low-limit setpoint temperature; this low-limit is the air temperature when the flow rate of outside air brought in is increased above the minimum airflow needed for ventilation. Researchers have not examined the low-limit's effect on energy conservation in great detail, even though basic airside economizers have been in use for many decades. This thesis provides an examination of the airside economizer's performance. It considers how low-limits affect energy consumption through an examination of different climate zones and a computational analysis. More importantly, this study provides a practical method for predicting the optimal low-limit of an airside economizer. Future research needed to improve the effectiveness of this control scheme is then suggested.

ACKNOWLEDGMENTS

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CHAPTER I

INTRODUCTION

The Industrial Revolution led many scientists and inventors to create new and to improve existing heating, ventilation, and air conditioning (HVAC) systems. In the present decade, the struggle is mainly to create more energy efficient HVAC equipment. Currently standards and codes require increasingly energy efficient HVAC systems, and that translates directly into energy savings for their users and reduced carbon emissions from most of the power plants that provide the needed electricity.

One HVAC device that has the potential for increasing energy efficiency in certain buildings and climates is the airside economizer. This is a control scheme intended to reduce cooling-mode energy consumption. When the outside air (OA) is cool, more air is introduced to replace or supplement the cooling and possibly the dehumidification that is provided by the cooling coil. For small commercial, institutional, and industrial (CII) buildings with packaged HVAC air handlers such as rooftop units (RTUs), adding simple dry-bulb temperature economizers with commonly-used fixed control setpoints will not produce optimal performance because airborne moisture is not considered. However, enthalpy controllers with multiple sensors have the ability to calculate the current heat balance, including the effects of moisture. They help to provide superior thermal performance but at a significantly increased initial cost; they typically need more maintenance as well. Dry-bulb control is thus more common; however, not monitoring the moisture introduced via the OA leads to increased energy consumption and periods of decreased thermal comfort as compared to enthalpy control.

An air-side economizer system has outdoor air, exhaust air, and recirculated air dampers to regulate these air flow rates. The dampers, and their attached ductwork, are sized to allow up to 100% of the required cooling supply airflow to be outside air. The recirculated air is the difference between the supply airflow and the exhaust airflow. When the OA is warm and humid the recirculated air damper is open to its maximum position, the exhaust air and OA dampers are in their minimum-for-ventilation-needs position. When the OA is cool and dry, the air-side economizer modulates the outdoor, exhaust, and recirculated air dampers, providing *all* necessary cooling via increased outside air flow, or to reduce the needed mechanically-provided cooling when the OA is of moderate temperature. Figure 1.1 is a schematic of an air-side economizer where:

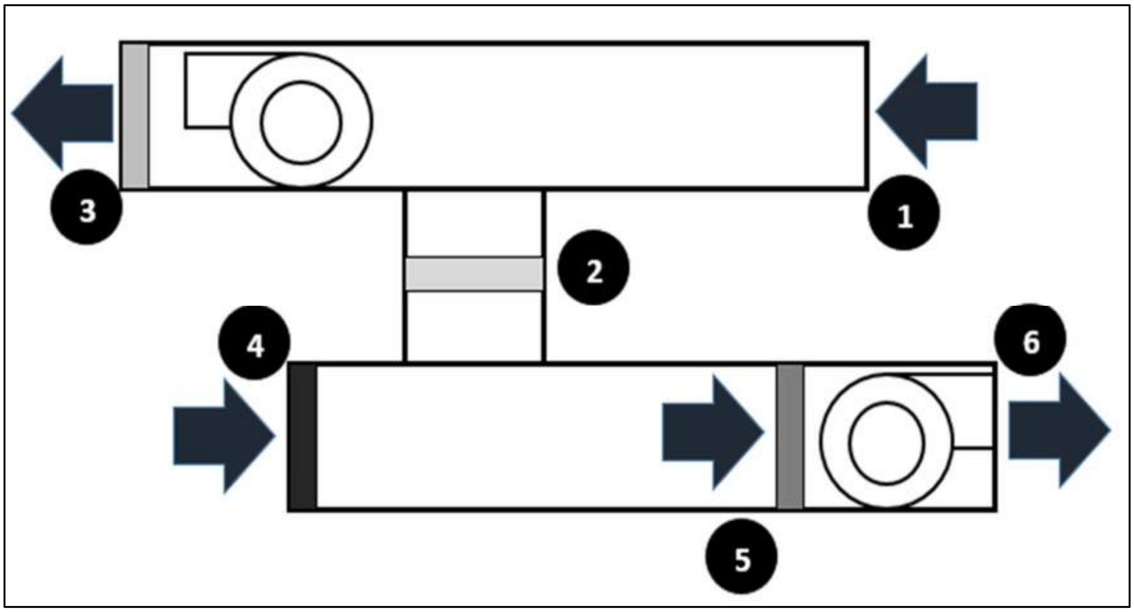

Figure 1.1 Air-side economizer system schematic

1. Return air (RA)
2. Recirculated air (CA) and its damper
3. Exhaust air (EA), its damper, and the EA fan
4. Outdoor air (OA) and its damper
5. Mixed air (MA) and air filter
6. Supply air (SA), and the SA fan

Figure 1.2 shows a drybulb temperature economizer control scheme, with the percentage of OA admitted varying with the temperature of the OA. Below the low-limit, the building is in space heating mode and only the minimum flow rate of outside air needed for ventilation is admitted. The low-limit OA temperature is at the point when the building should switch from heating to cooling mode. With an air-side economizer, as the cooling load increases the interlinked OA, CA, and EA dampers modulate to maintain the temperature of the indoor spaces; without the economizer the minimum OA would continue to be admitted but the mechanical cooling system would be activated. But with the drybulb airside economizer, above the low-limit, as shown in Figure 1.2, the percentage of OA increases linearly with the temperature of the OA. The system is in “free”-cooling mode, until it reaches the middle-limit where there is 100% OA and the temperature of the OA is typically 55°F (12.8°C), the common design supply air temperature during cooling mode. As the outdoor temperature increases above the middle-limit, the OA remains constant at 100% but the mechanical cooling is activated to keep the supply air at 55°F (12.8°C). Once the temperature of the OA reaches the high-limit, the percentage of OA is reduced to the minimum-ventilation required once again to minimize mechanical cooling energy consumption. This minimum percent OA is typically 10 to 20% for code-compliant office buildings in the United States.

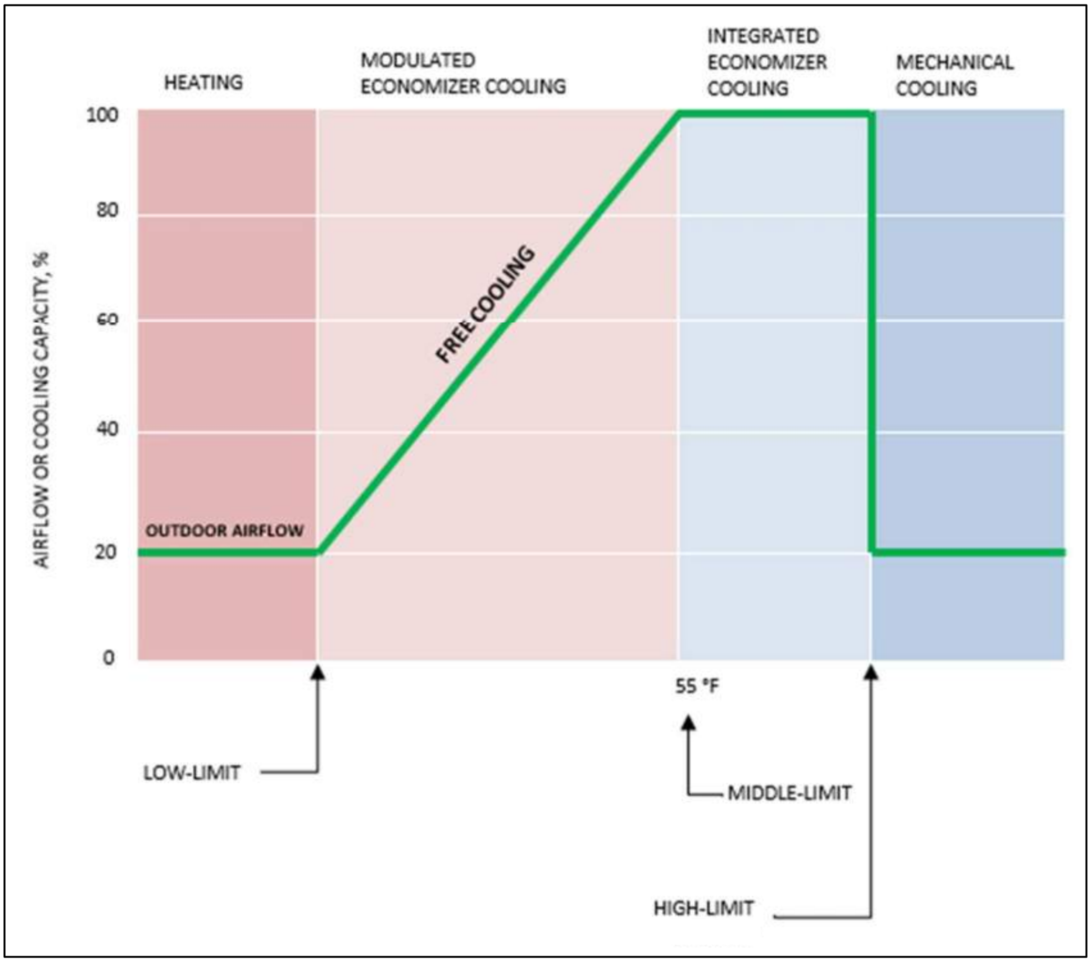

Figure 1.2 Percent Outside air versus Drybulb Temperature control scheme

For these simple, widely-used dry-bulb controllers, the three OA temperature setpoints of concern are thus 1) the low-limit, where need for indoor cooling begins, 2) full "free-cooling" middle-limit where the mixed air temperature matches the design cooling-mode supply air temperature, and 3) the high-limit where the percent OA is decreased to the minimum because the OA is too warm or humid. Other researchers have studied the high-limit in detail, e.g., Taylor and Cheng 2010, and have produced practical advice for selecting it. The middle-limit is the cooling mode design supply air temperature as specified by the design engineer and is often 55°F (12.8°C) for conventional HVAC systems in North

America.

Researchers have not previously focused on the effect of low-limit setpoints. The low-limit setpoint is problematic, as it is normally assumed equal to the building's balance point temperature, which is an unknown. However, due to lack of knowledge as to what that low-limit should be set to, the installer will often guess the balance point temperature. As a result, energy consumption will be higher than optimal either by over- or under-cooling the building unless the installer makes an excellent estimate of the optimal setpoint.

The purpose of this research study was to examine the effect of dry-bulb low-limits on the annual cooling energy consumption of an office building in various climates. While the high-limit is discussed, it was not the focus of this study.

An existing occupied building was selected as the base case for the study for realism and because actual energy usage records were obtainable. In a later chapter the base case will be discussed in detail. The building's annual energy performance, without utilizing an airside economizer, was then simulated using an hour-by-hour computational model for different climates of the United States. By doing so the balance point temperature could be observed. Then the simulations were performed again but with the airside economizer activated and the low-limits set to these balance point temperatures in each of the different climate zones. In addition, the low-limit was varied through a range of other temperatures thus allowing for the comparison of energy savings to observe the optimal values. However, before the simulations were performed, a literature review was conducted, and its results appear in Chapter II of this thesis.

CHAPTER II

LITERATURE REVIEW

New mid- to large-sized buildings' HVAC systems often employ air- or water-side economizers. While economizers have been around for decades, it is within the last ten years or so that the use of an economizer was strongly suggested or even outright required by building energy conservation codes such as ANSI/ASHRAE Standard 90.1 (ASHRAE 90.1-2013). The benefit of the airside economizer is that free- or reduced-cost cooling can be accomplished similarly to natural ventilation schemes but without the problems of natural ventilation such as less-than-optimal user operation of windows (Emmerich et al. 2003).

Different types of air-side economizer control strategies exist. Types of airside economizers include fixed drybulb, differential dry bulb, fixed enthalpy, electronic enthalpy, differential enthalpy, and dewpoint-and-drybulb. The fixed drybulb is simple: when the outside air (OA) increases to a fixed temperature of about 68°F (20°C) to 72°F (22.2°C), the economizer is disabled so that dampers return to the minimum position needed for admitting ventilation air. With differential drybulb, the economizer becomes disabled when the OA is warmer than the return air. Similar to fixed drybulb, the fixed enthalpy type is disabled when the OA reaches a fixed enthalpy. When the OA reaches a predefined high-limit drybulb and dewpoint, the electronic enthalpy type disables the economizer. In the case of differential enthalpy, an economizer will be disabled when the return air enthalpy is less than the OA enthalpy. Once achieving the desired fixed dewpoint or drybulb, the dewpoint-and-drybulb type reduces the percent OA. To achieve the goal of uniform compliance with ASHRAE Standard 90.1, one of these economizers is often required. Before this code mandate, the airside economizer control scheme selected, if any, was dependent on the desired initial cost of the system and the predicted operating costs (Trane 2006).

According to Taylor et al. (2010), each of the different types of control systems and their schemes introduce errors. Their study showed this causes an increased use of energy when compared to theoretically perfect control logic and performance. Using the different climate zones provided in ASHRAE 90.1, they developed a table of proposed high limits. Taylor et al. (2010) gives these high limits based upon device type, climate zone, and high-limit logic. Also included in this table are the device types that would not be recommended for each climate zone. Table 2.1 displays these values.

Table 2.1 High-limit control for integrated economizers [Taylor et al. 2010]

Device Type	Acceptable in Climate Zone at Listed Setpoint	High Limit Logic (Economizer Off When):		Not Recommended In Climate Zone
		Equation	Description	
Fixed Dry Bulb	3C, 6B, 8	$T_{OA} > 75°F$	Outdoor air tempurature exceeds 75°F	
	1B, 2B, 3B, 4B, 4C, 5B	$T_{OA} > 73°F$	Outdoor air tempurature exceeds 73°F	
	5C, 6A, 7	$T_{OA} > 71°F$	Outdoor air tempurature exceeds 71°F	
	1A, 2A, 3A, 4A, 5A	$T_{OA} > 69°F$	Outdoor air tempurature exceeds 69°F	
Differential Dry Bulb	1B, 2B, 3B, 3C, 4B, 4C, 5B, 5C, 6B, 7, 8	$T_{OA} > T_{RA}$	Outdoor air tempurature exceeds return air tempurature	1A, 2A, 3A, 4A, 5A, 6A
Fixed Enthalpy	4A, 5A, 6A, 7, 8	$h_{OA} > 28$ Btu/lb*	Outdoor air enthalpy exceeds 28 Btu/lb of dry air*	All
Fixed Enthalpy + Fixed Dry Bulb	All	$h_{OA} > 28$ Btu/lb* or $T_{OA} > 75°F$	Outdoor air enthalpy exceeds 28 Btu/lb of dry air* or outdoor air dry bulb exceeds 75°F	All
Electronic Enthalpy	All	$(T_{OA}, RH_{OA}) > A$	Outdoor air tempurature/RH exceeds the "A" setpoint curve.†	All
Differential Enthalpy	None	$h_{OA} > h_{RA}$	Outdoor air enthalpy exceeds return air enthalpy.	All
Differential Enthalpy + Fixed (or Differential) Dry Bulb	None	$h_{OA} > H_{RA}$ or $T_{OA} > 75°F$ (or T_{RA})	Outdoor air enthalpy exceeds return air enthalpy or outdoor air dry bulb exceeds 75°F (or return air tempurature)	All
Dew Point + Dry-Bulb Tempuratures	None	$DP_{OA} > 55°F$ or $T_{OA} > 75°F$	Outdoor dew point exceeds 55°F (65 gr/lb) or outside air dry bulb exceeds 75°F	All

* At altitudes substantially different than sea level, the fixed enthalpy limit shall be set to the enthalpy value at 75°F and 50% relative humidity. As an example, at approximately 6,000 ft elevation the fixed enthalpy limit is approximately 30.7 Btu/lb

† Setpoint "A" corresponds to a curve on the psychrometric chart that goes through a point at approximately 73°F and 50% relative humidity and is nearly parallel to dry-bulb lines at low humidity levels and nearly parallel to enthalpy lines at hugh humidity levels.

Another concern associated with air-side economizer control is that building pressurization can affect adversely its performance. If the HVAC system brings in OA without providing a way for the extra air to escape, the building will develop excess indoor air pressure, have difficulty in bringing in more OA, create moisture concerns in the building enclosure, and have troubles with exterior doors' operation. Besides increasing the ducted exhaust airflow rate, ways exist to reduce this excess building pressure, one of which is the use of local barometric relief dampers that are also known as gravity dampers. When using an air-side economizer device some method of building pressure control must be implemented (Trane 2006).

A constant-air-volume (CAV) system control scheme for an air-side economizer is slightly different than the control scheme for a variable-air-volume (VAV) system. In cold weather, the heating load of the admitted OA decreases as the OA temperature rises toward the low-limit. The overall heating load is the heat rate added to a space to maintain the building temperature; the cold OA that's admitted is a portion of this heating load. To minimize energy consumption, the minimum OA airflow needed for ventilation enters the system when in heating mode. Above the low-limit, the building's load changes from heating to cooling. Even if mechanical cooling isn't provided, above the low-limit the ventilation system can enter modulated or "free-cooling" economizer mode, where the minimum then up to 100% OA is admitted to provide sufficient cooling until the middle-limit is reached. With CAV, the OA and recirculated air flow rates modulate inversely to maintain the space temperature as the building cooling load varies; the total SA flowrate remains the same. In buildings with mechanical air-conditioning, above the middle-limit that mechanical cooling and dehumidification is increasingly employed but 100% OA is maintained until the high-limit is reached because the OA's enthalpy is still hopefully below that of the return air. In Figure 2.1, the dark grey area represents the mechanical cooling energy saved when the system is in modulated economizer mode as compared to cooling with just the minimum OA needed for ventilation. Between the low- and middle-limits, the OA damper gradually opens from minimum to 100% as the cooling load increases. As this occurs, the recirculated air-damper gradually closes. The system is in integrated economizer mode when 100% OA is providing part of the cooling capacity necessary. When the building is in an unoccupied period, small- to mid-sized mechanical cooling systems often cycle on/off as needed to maintain the temperature within the space. In Figure 2.1, the white area represents the energy savings when the economizer is in integrated economizer mode. After reaching the high-limit, the integrated economizer mode will deactivate; the system will then return to mechanical cooling-only, with the minimum required OA flow for ventilation (Trane 2006).

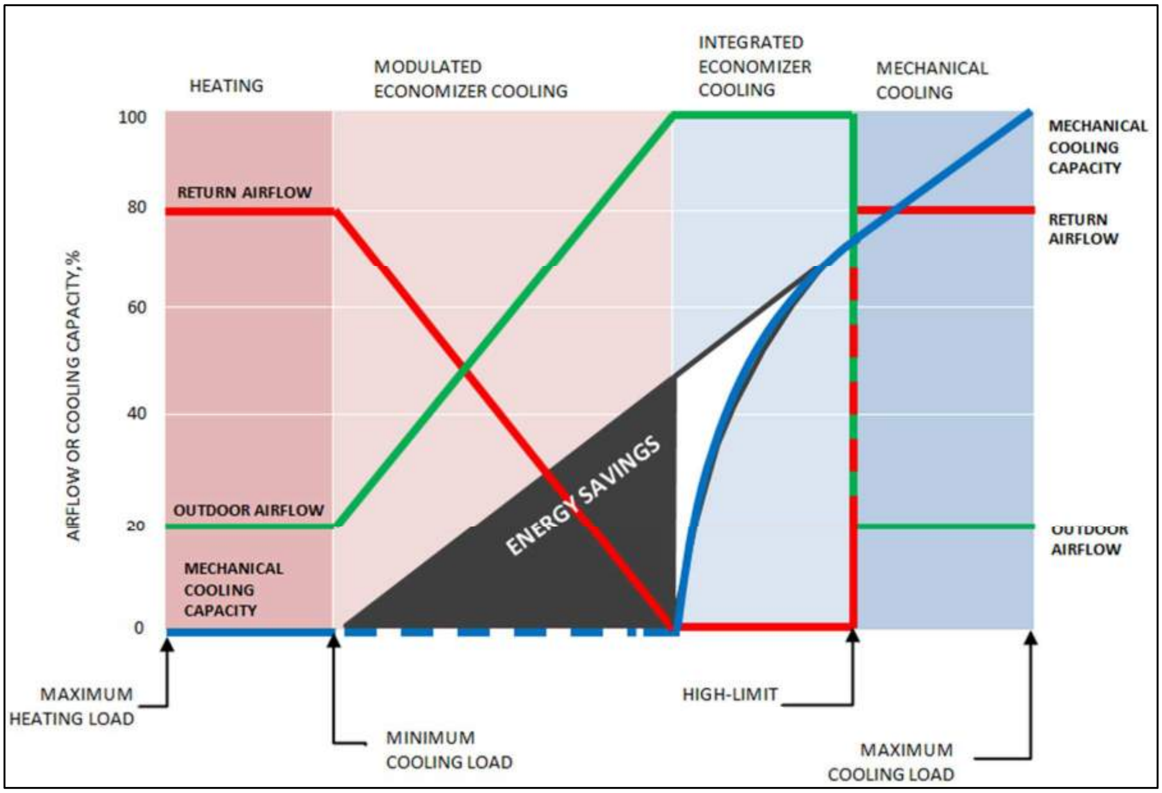

Figure 2.1 Potential savings provided via a typical economizer control scheme for a constant-air-volume (CAV) system (Trane 2006).

In heating mode the temperature difference between the supply air (SA) and the indoor setpoint can be very high. As a result, with VAV instead of CAV, the supply air flow rate is likely far less than the maximum supply air flow rate typical for the design cooling-load conditions when the design room-to-supply air temperature difference is much smaller than for heating; in VAV systems the maximum heating SA flow rate is often only about half that of the maximum cooling flow rate. Modulated economizer mode begins at the low-limit when the cooling load begins to increase, but with VAV only the OA flow rate increases, not necessarily the recirculated air, so less air may need to be moved. As with CAV, during integrated economizer mode between the middle and high limits, the outdoor damper is 100% open, and the cooling coil is active. Fans and dampers are modulated to provide an air pressure balance. As with CAV, the system will enter mechanical cooling with minimum required OA mode after having reached the high-limit as seen in Figure 2.2. With VAV, the supply and return airflow rates increase to meet higher cooling loads, and the cooling coil provides the necessary cooling and dehumidification capacity.

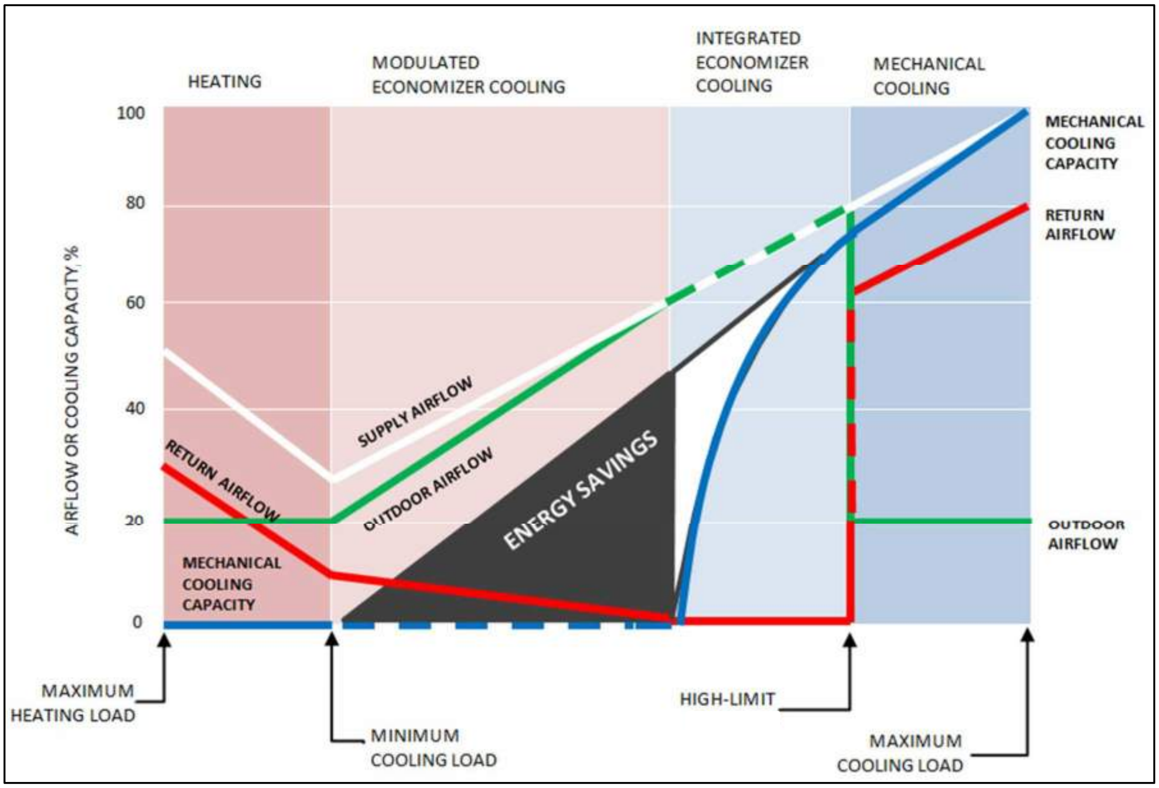

Figure 2.2 Potential savings for a typical economizer control scheme for variable-air-volume (VAV) systems (Trane 2006).

Another concern with air-side economizers is sensor placement because the sensors' readings will greatly affect the performance of the control scheme. The placement can affect the operation and maintenance costs too. In addition to the sensor placement, the type of sensor is important as well. As previously stated, a variety of controls and sensors are available.

Properly calibrated sensors are important. ASHRAE Standard 90.1-2013 contains a section on required sensor accuracy for air-side economizers [ASHRAE 90.1-2013, §6.5.1.1.6]:

Outdoor air, return air, mixed air, and supply air sensors shall

be calibrated within the following accuracies: a. dry-bulb temperatures shall be accurate to ±2°F over the range of 40°F to 80°F; b. enthalpy and the value of a differential enthalpy sensor shall be accurate to ±3 Btu/lb over the range of 20 to 36 Btu/lb; c. relative humidity shall be accurate to ±5% over the range of 20% to 80% RH.

From this section of ASHRAE Standard 90.1-2013, it is easy to see how accurate a system must be, because as previously noted a small difference in sensor readings can greatly affect the accuracy and thus the efficiency of the system.

One of the requirements in ASHRAE Standard 90.1 in regards to economizers is that the mechanical cooling systems must integrate the economizer system; "integration" means that the economizer system becomes a part of the mechanical cooling system. According to Standard 90.1-2013, §6.5.1.3,

Economizer systems shall be integrated with the mechanical cooling system and shall be capable of providing partial cooling even when additional mechanical cooling is required to meet the remainder of the cooling load. Controls shall not be a false load on the mechanical cooling systems by limiting or disabling the economizer or by any other means, such as hot gas bypass, except at the lowest stage of mechanical cooling.

When sizing an air-side economizer system, the system must have the appropriate design capacity. ASHRAE Standard 90.1-2013 states, "Air economizer systems shall be capable of modulating OA and return air dampers to provide up to 100% of the design supply air quantity as OA for cooling."

This requirement continues in Section 6.5.1, Economizers, of Standard 90.1-2013, which states that "Each cooling system that has a fan shall include either an air or water economizer meeting the requirements of sections 6.5.1.1 through 6.5.1.6." The exceptions are [ASHRAE 90.1-2013, §6.5.1]:

1. Individual fan-cooling units with a supply capacity less than the minimum listed in Table 6.5.1-1 for comfort cooling applications and Table 6.5.1-2 for computer room applications.
2. Systems that include nonparticulate air treatment as required by Section 6.2.1 in Standard 62.1.
3. In hospitals and ambulatory surgery centers, where more than 75% of the air designed to be supplied by the system is to spaces that are required to be humidified above 35°F dew-point temperature to comply with applicable codes or accreditation standards; in all other buildings, where more than 25% of the air designed to be humidified above 35°F dew-point temperature to satisfy process needs. This exception does not apply to computer rooms.
4. Systems that include a condenser heat recovery system with a minimum capacity as defined in Section 6.5.6.2.2.
5. Systems that serve residential spaces where the system capacity is less than five times the requirement listed in Table 6.5.1-1.
6. Systems that serve spaces whose sensible cooling load at design conditions, excluding transmission and infiltration loads, is less than or equal to transmission and infiltration losses at an outdoor temperature of 60°F.
7. Systems expected to operate less than 20 hours per week.
8. Where the use of outdoor air for cooling will affect supermarket open refrigerated casework systems.
9. For comfort cooling where the cooling efficiency meets or exceeds the efficiency improvement requirements in Table 6.5.1-3.
10. Systems primarily serving computer rooms where
   a. the total design cooling load of all computer rooms in the building is less than 3,000,000 Btu/h and the building in which they are located is not served by a centralized chilled water plant;
   b. the room total design cooling load is less than 600,000 Btu/h and the building in which they are located is served by a centralized chilled water plant;
   c. the local water authority does not allow cooling towers; or
   d. less than 600,000 Btu/h of computer-room

cooling equipment capacity is being added to an existing building

11. **Dedicated systems for computer rooms where a minimum of 75% of the design load serves**
    a. **those spaces classified as an essential facility,**
    b. **those spaces having a design of Tier IV as defined by ANSI/TIA-942,**
    c. **those spaces classified under NFPA 70 Article 708 – Critical Operations Power Systems (COPS), or**
    d. **those spaces where core clearing and settlement services are performed such that their failure to settle pending financial transactions could present systemic risk as described in "The Interagency Paper on Sound Practices to Strengthen the Resilience of the U.S. Financial System, April 7, 2003"**

The tables mentioned in this list of exceptions are repeated in this thesis as Tables 2.2 through 2.4. Figure 2.3, from ASHRAE 90.1 is an adapted climate zone map.

Table 2.2 Adapted from ASHRAE Standard 90.1-2013 Table 6.5.1-1

Minimum Fan-Cooling Unit Size for which an Economizer is Required for Comfort Cooling	
Climate Zones	**Cooling Capacity for which an Economizer is Required**
1a, 1b	No economizer requirement
1a, 2b, 3a, 4a, 5a, 6a, 3b, 3c, 4b, 4c, 5b, 5c, 6b, 7, 8	≥54,000 Btu/h

Table 2.3 Adapted from ASHRAE Standard 90.1-2013 Table 6.5.1-2

Minimum Fan-Cooling Unit Size for which an Economizer is Required for Comfort Cooling	
Climate Zones	**Cooling Capacity for which an Economizer is Required**
1a, 1b	No economizer requirement
1a, 2b, 3a, 4a, 5a, 6a, 3b, 3c, 4b, 4c, 5b, 5c, 6b, 7, 8	≥54,000 Btu/h

Table 2.4 Adapted from ASHRAE Standard 90.1-2013 Table 6.5.1-3

Eliminate Required Economizer for Comfort Cooling by Increasing Cooling Efficiency	
Climate Zone	**Efficiency Improvement [a]**
2a	17%
2b	21%
3a	27%
3b	32%
3c	65%
4a	42%
4b	49%
4c	64%
5a	49%
5b	59%
5c	74%
6a	56%
6b	65%
7	72%
8	77%

a. If a unit is rated with an IPLV, IEER, or SEER then to eliminate the required air or water economizer, the minimum cooling efficiency of the HVAC unit must be increased by the percentage shown. If the HVAC unit is only rated with a full-load metric like EER cooling then these must be increased by the percentage shown

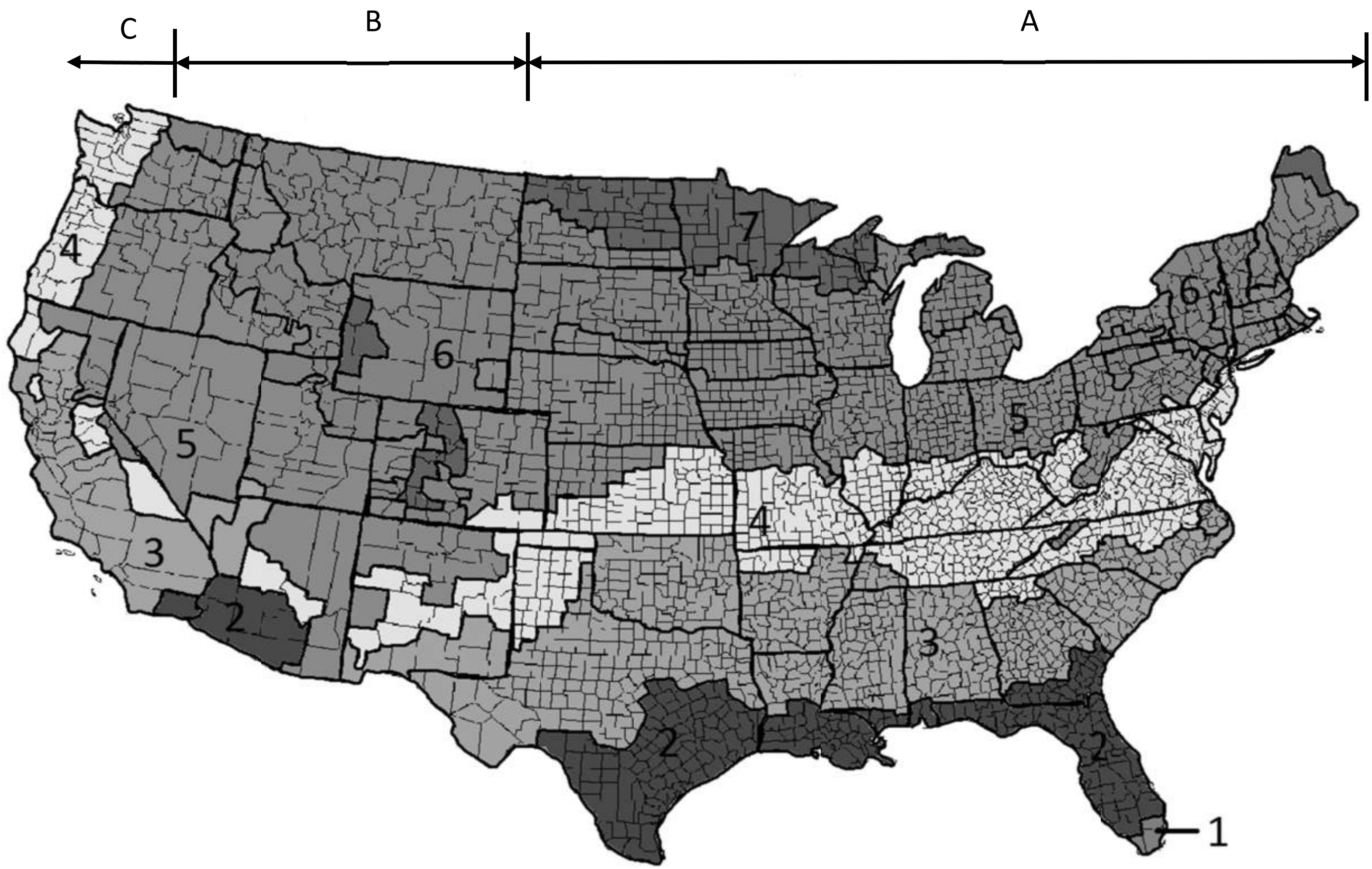

Figure 2.3 Adapted from ASHRAE 90.1 Climate Map

Minimum ventilation air requirements for commercial and institutional buildings are defined in ASHRAE Standard 62.1. That minimum is "The outdoor air flow required in the breathing zone of the occupiable space or spaces in a ventilation zone, i.e., breathing zone outdoor air flow (V_{bs}), shall be no less than the value determined in accordance with Equation 6.2.2.1" (ASHRAE 62.1-2013). Equation 6.2.2.1 of the standard is shown as Equation 2.1 of this thesis.

$$\boldsymbol{V_{bz} = R_p \text{ x } P_z + R_a \text{ x } A_z} \qquad \textbf{(2.1)}$$

where

A_z represents the floor area of the zone, ft^2 (m^2)

P_z is the population of the zone, number of people

R_p is the outside airflow rate per person, and

R_a is the outside airflow rate per unit floor area.

Values for both R_p and R_a are given in Table 6.2.2.1 of Standard 62.1-2013 (ASHRAE 2013).

The design engineer for such a system will need to examine the various operational conditions over the entire year. For a data processing center with no economizer, for example, the cooling system might need to run all 8,760 hours per year to remove the heat released by the many continuously-operating computers and networking devices within. Due to this, the potential for savings via the use of an air-side economizer is great, depending on the climate and other factors.

As such, one of the most common applications of economizers, aside from office buildings, is in data centers. Data centers are a major energy consumer. The concerns with airside economizers' use in data centers pertain to outdoor particulates and other contaminants making their way into the building, as well as the ability to control the humidity within the spaces served. Many of the concerns relating to OA in data centers are being addressed in ASHRAE's Technical Committee (TC) 9.9. While there is apprehension, more engineers and designers are using airside economizers in data centers, and data centers are often intentionally being built in cold climates rather than warm.

Most recently in Santa Clara, California, in Building 4 of the Marvell Semiconductor U.S. headquarters, an airside economizer was installed on an existing data center; Santa Clara has a mild, but not cold climate. These project retrofitted airside economizers on to the already installed computer room air handlers (CRAHs). With an uninterruptible power supply added too as part of the project, the cost was approximately US$662,000. Once the project was completed and the economizers were functioning properly, the City of Santa Clara's electrical utility awarded a rebate of US$171,000. Before the completion of the project, the electrical energy consumption for the 12 months prior was approximately 1,361,450 kWh per month. After completion of the project, the building was, on average, consuming 1,091,280 kWh per month, resulting in a 270,170 kWh reduction per month. This was a 20% reduction in the overall building electrical energy consumption and 30% in the data center. With US$27,000 per month savings in energy costs, the simple payback was only 18 months [Alipour 2013].

Despite the large potential energy savings through the use of airside economizers, when properly specified, installed, and operated, many such control schemes never achieve their design intent. Identifying and then using optimal setpoints is essential in achieving the desired energy savings, and best thermal comfort.

CHAPTER III

BASE CASE

The commercial building used in this study is in Lenexa, Kansas at 9701 Renner Boulevard. In 2006, the building was completed and occupancy of it began. The gross building area is 129,321 square feet, and the net usable area is 108,096 square feet, with four floors to the building. Trees and shaded areas surrounding the building are sparse and offer little to no shading on the building's exterior as is shown in Figure 3.1

Figure 3.1 The building used for this study was 9701 Renner Blvd, Lenexa, KS.

Building Construction

The building has a large number of windows. The glazings in the building are double-pane 0.2362 inch (6 mm) tinted glass. The U-value of the windows are 0.505 $Btu/h \cdot ft^2 \cdot °F$ (2.87 $W/m^2 \cdot K$). The exterior walls have precast concrete on their outside and standard gypsum wallboard on the inside. Due to not having the exact composition of the construction materials on the inside of the walls, the overall heat transfer coefficient (U-value) of the wall assembly was assumed to be typical for this type of building at approximately 0.0526 $Btu/h \cdot ft^2 \cdot °F$ (0.297 $W/m^2 \cdot K$). Similarly, the U-value of the roof assembly was assumed to be approximately 0.0333 $Btu/h \cdot ft^2 \cdot °F$ (0.189 $W/m^2 \cdot K$). The floor slab of the building is assumed to be 8 inch thick (0.2032 m) heavyweight concrete.

HVAC System

For typical large United States commercial office buildings the most common HVAC system used is variable-air-volume (VAV) with reheat; the base case building uses such and is also all-electric as are many such buildings; no natural gas is utilized. Each floor of the building was split by its HVAC designer into multiple thermal zones. As such, in this study, the base case included an all-air HVAC system with VAV terminal units with electric resistance reheat serving the many thermal zones. To establish the base energy consumption per year each of the first group of simulations had the same system with only the geographical location of the building changed. In the second set of simulations, the airside economizer was activated. Initially these airside economizer cases kept the low-, middle-, and high-limit set points of the economizer as the defaults of the simulation program; later the low-limit was varied.

The actual building's monthly electricity use was recorded by a facilities manager from the January 2006 to through November 2012. In 2012, the overall annual energy consumption was lower than the average for the six years prior. From December 2012 to 2013 the building was not fully occupied, so those months' data were not utilized for this study. Table 3.1 shows the monthly electrical energy use of the building.

Table 3.1 Electricity Usage from 2006 to 2012 (Courtesy of Kiewit)

Electricity Usage							
kWh							
	2006	**2007**	**2008**	**2009**	**2010**	**2011**	**2012**
JAN	304,800	336,800	322,800	328,000	303,200	331,200	275,200
FEB	346,400	366,000	386,800	296,000	295,600	304,400	290,400
MAR	277,200	253,200	328,000	249,600	252,400	252,800	211,200
APR	269,600	280,000	288,800	266,400	238,800	250,800	217,200
MAY	272,000	267,600	276,400	239,600	226,400	232,000	214,200
JUN	270,800	283,600	268,800	248,800	233,600	236,800	234,000
JUL	322,000	294,000	276,800	251,600	261,200	285,600	216,400
AUG	282,400	290,800	283,200	217,200	246,000	233,600	187,200
SEP	270,400	286,800	292,000	213,200	224,000	222,800	197,600
OCT	282,000	256,800	266,000	222,400	220,800	230,000	200,000
NOV	318,000	334,400	324,000	263,200	308,400	275,200	232,400
DEC	336,800	365,600	353,200	352,000	364,000	315,600	-
Total	3,552,400	3,615,600	3,666,800	3,148,000	3,174,400	3,170,800	2,475,800
Average	296,033	301,300	305,567	262,333	264,533	264,233	206,317

As previously noted, the building has four levels, and they are the Lower Level, Floor 1, Floor 2, and Floor 3. The Lower Level's footprint is only the south half of the building. This floor is partially underground due to the site's sloping grade. Floors 1 through 3 cover the entire footprint of the building. Private offices comprise 26.6% of the building's total floor area. Open-plan cubicles utilize another 24.4% of the building. The building's floor plans appear in Figures 3.2 to 3.5.

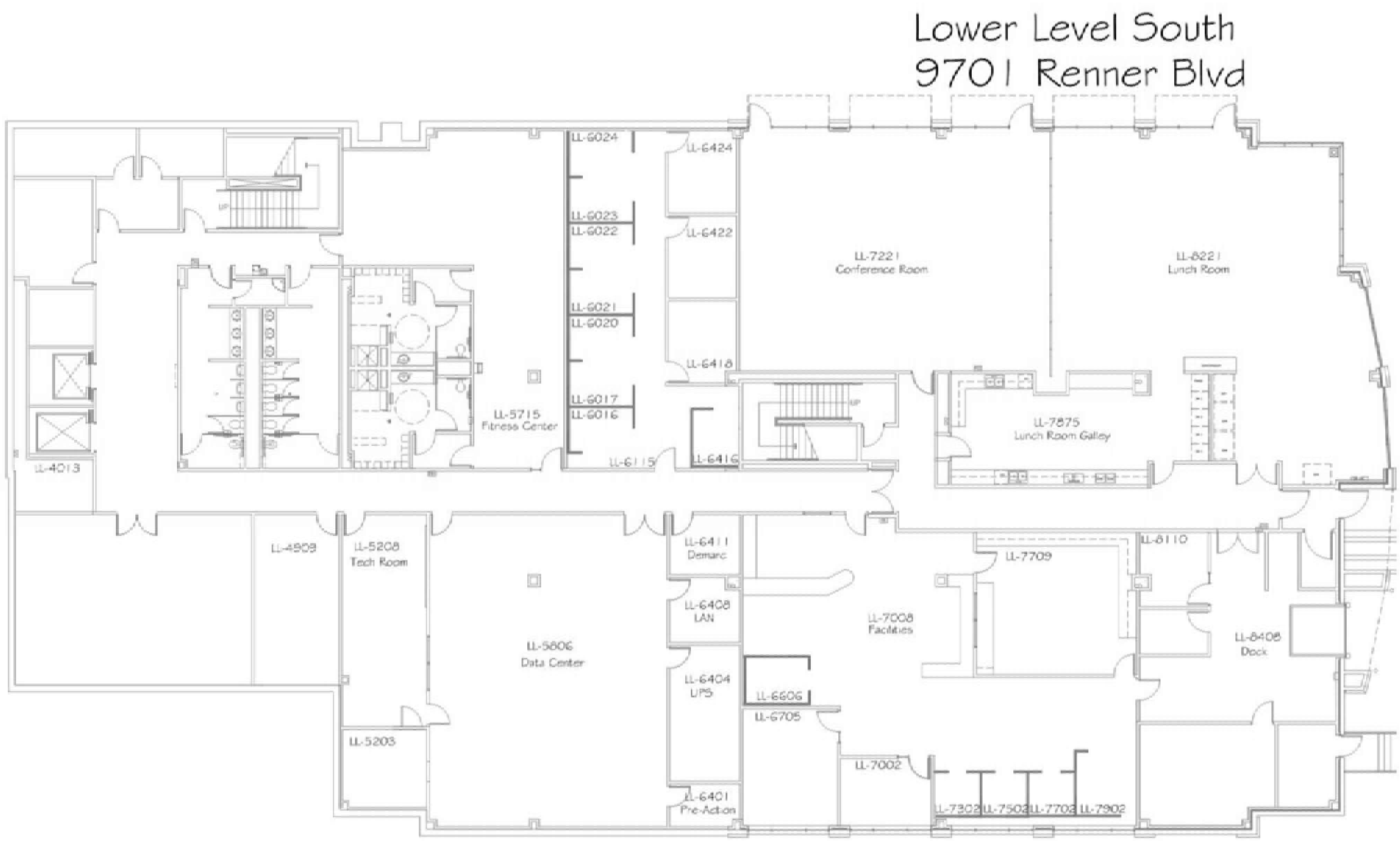

Figure 3.2 9701 Renner Blvd Lower Level Floor Plan

Figure 3.3 9701 Renner Blvd 1st Level Floor Plan

Figure 3.4 9701 Renner Blvd 2nd Level Floor Plan

Figure 3.5: 9701 Renner Blvd 3rd Level Floor Plan

HVAC Load and Energy Analysis Software Utilized

Two separate modeling programs were used to perform the needed thermal analyses of the building. Trane'sTrace^{TM}700 was used to perform the initial calculations (Trane 2013). The then-latest version, 6.3, was used. Complete with subroutines in load design, system simulation, equipment simulation, energy consumption and economic analyses, Trace^{TM}700 was deemed, at least at first, a logical program for modeling the building and the proposed HVAC system modifications.

Trace^{TM}700 and all other building energy simulation programs require much input data. Building descriptions that include location, zones, materials, and weather data are needed by the load and energy calculations subroutines. Each of the HVAC systems' input data sets are comprised of system type, temperature and humidity setpoints, economizer type, and dedicated OA scheme, for example. Peak and hourly loads are determined by the load-design subroutine. Airflow and supply air temperatures by zone are extracted from the load-design subroutine's results and then used by the system simulation subroutine. Both the load subroutine and the system simulation subroutine use data from a weather library. From the system simulation subroutine, the hour-by-hour equipment loads are determined and then utilized further in the equipment simulation subroutine. Transient equipment' performance descriptions such as pump and fan curves are included in the equipment simulation subroutine via a performance library. After determining each piece of equipment's energy consumption, and then summing them for each primary energy source, the economic analysis is performed by the program. However, the economic portion of the software was not needed for this research project, only the predicted annual energy use for each case.

The second modeling software used was eQUEST; it is a user-friendly shell for DOE-

2 (eQuest 1999). This software was used strictly for energy modeling purposes. Version 3.65 was the most-current version of eQUEST and was utilized for this study. eQUEST is short for "The Quick Energy Simulation Tool." The shell program uses text-entry windows in which the users are led through steps to define the building. Based on the initial choices made within the software's early input windows, there would be either more or fewer steps taken. Through further steps, eQUEST allows the user to run simulations and then the results are provided in user-friendly formats such as graphs and tables instead of the raw data files of DOE-2 (eQuest 1999).

Software Inputs

Discrepancies can occur when using multiple software programs. It is important when using different types of software that the input data match, which can be very difficult sometimes. With Trace™700 and eQUEST, the data entry style into the programs is different, and the programs' default assumptions are not necessarily the same. The biggest instance was the dimensions. For example, in Trace™700, each space within the building is input separately. In eQUEST the buildings outer perimeter is entered and then the internal spaces are all described by percentages.

The overall exterior wall area, including windows for the building is approximately 30,900 square feet. Of that area, the wall area that is glass is 60%. The solar load through all of this glass is fairly high and does not differ much in the predicted annual peak loads for the various sites used in the United States due to a small range of latitudes.

Similarly to glass, infiltration is an envelope load. In both Trace™700 as well as eQUEST, the chosen value for infiltration was 0.3 air changes per hour (ACH). This value for infiltration applies to the perimeter of the building and is a design-estimate for a commercial building with neutral pressurization.

Similarly, ventilation must be taken into account. Ventilation, as defined by ASHRAE Standard 62.1-2013, is "the process of supplying air to or removing air from a space for the purpose of controlling air contaminant level, humidity, or temperature within the space" (ASHRAE 62.1-2013). The ventilation air flow rate is determined using many factors such as number of occupants, occupancy type, floor area, and ventilation system effectiveness. The following ventilation air flow rates were found from ASHRAE Standard 62.1-2013, and the majority of spaces resulted in 15 CFM per person. But the fitness center needed 20 CFM per person, the restrooms required 10 CFM per person, and the mechanical/electrical/storage spaces resulted in 0.06 CFM per square foot. Due to the ventilation rate depending on occupancy in most spaces, also needed were the estimated occupancy for each space. For the offices, that were the majority of spaces of the building, the occupancy was 143 square feet per person as recommended in ASHRAE Standard 62.1. Other spaces, such as the fitness center and the mechanical/electrical/storage spaces, had different densities. The fitness center had 17 square feet per person and the mechanical/electrical/storage spaces had no occupants. These maximum occupancies were also used to determine the sensible and latent heat loads of the people, which are significant internal cooling loads of the building.

Occupancy loads are not the only internal loads; lighting and equipment loads were also to be modeled. The Fundamentals volume of the ASHRAE Handbook (ASHRAE 2013) has a table of lighting's watts per square feet, as well as for various types of office equipment. The values used appear in Table 3.2.

Table 3.2 Lighting, office, and miscellaneous loads (ASHRAE 2013)

Activity Space	Lighting (W/sqft)	Office Equipment (W/sqft)	Miscellaneous Equipment (W/sqft)
Cubicals	1.1	0.5	0.0
Conference Room	1.3	1.0	1.0
Office	1.1	1.0	0.0
Breakroom	1.3	0.0	1.0
Mechanical/Electrical/Storage	1.5	0.0	10.0
Lobby	1.1	0.0	0.0
Fitness Center	0.9	0.0	1.0
Restrooms	1.1	0.0	0.0

In Trace^{TM}700, the central cooling equipment chosen were air-cooled vapor compression unitary RTUs, with electric resistance heating coils. In eQUEST, similar to Trace^{TM}700, the heating source chosen was thus electric resistance. Within both of the programs, a standard variable air volume (VAV) with electric-resistance reheat air distribution system was chosen; the VAV minimum flow was 30% in both models. Also similar supply and return fan data were specified within both programs. For both the supply and return, forward-curved centrifugal fans with variable frequency drives were selected.

Through use of the software and the aforementioned inputs and values, the programs determined results for the base case in the various climates. Examples of some spaces' input for Trace^{TM}700 can be found in the Appendix.

CHAPTER IV

BASE CASE PREDICTIONS BY CLIMATE ZONE

Climate zones are simply averaged geographical divisions; each zone is different based on a variety of weather factors. To improve the accuracy of results, ASHRAE Standard 90's committee has divided each climate zone into sub-zones A, B, and C. Sub-zone A signifies a more moist environment, B represents a drier environment, and C is representative of a marine environment. For example, greater-San Diego's weather ranges from mariene along its bay to hot and dry desert just to its east. Ideally, the environment for an airside economizer to function at its highest effectiveness is when the building needs cooling, and the available OA is cool and its relative humidity (RH) is low. A previous study by Taylor et.al. Cheng, examined the effects of economizer high-limits and provided good choices for locations for this current study of the low-limit. Table 4.1 lists these various locations.

Table 4.1 Locations selected within each North American climate zone

Climate Zone	Location
1A	Miami, FL
2A	Houston, TX
2B	Phoenix, AZ
3A	Atlanta, GA
3B	Los Angeles, CA
3C	San Francisco, CA
4A	Baltimore, MD
4B	Albuquerque, NM
4C	Seattle, WA
5A	Chicago, IL
5B	Boulder, CO
5C	Vancouver, BC
6A	Minneapolis, MN
6B	Helena, MT
7	Duluth, MN
8	Fairbanks, AK

Base Case's Annual Energy Consumption Predictions

The first step in determining the optimal low-limits was to find the base annual energy consumption for each of the 16 climate zones when no air-side economizer was utilized. These values gave the basis for finding the savings potential of the air side economizer as the low-limit was varied. The results for each of the different climate zones appear in Figure 4.1.

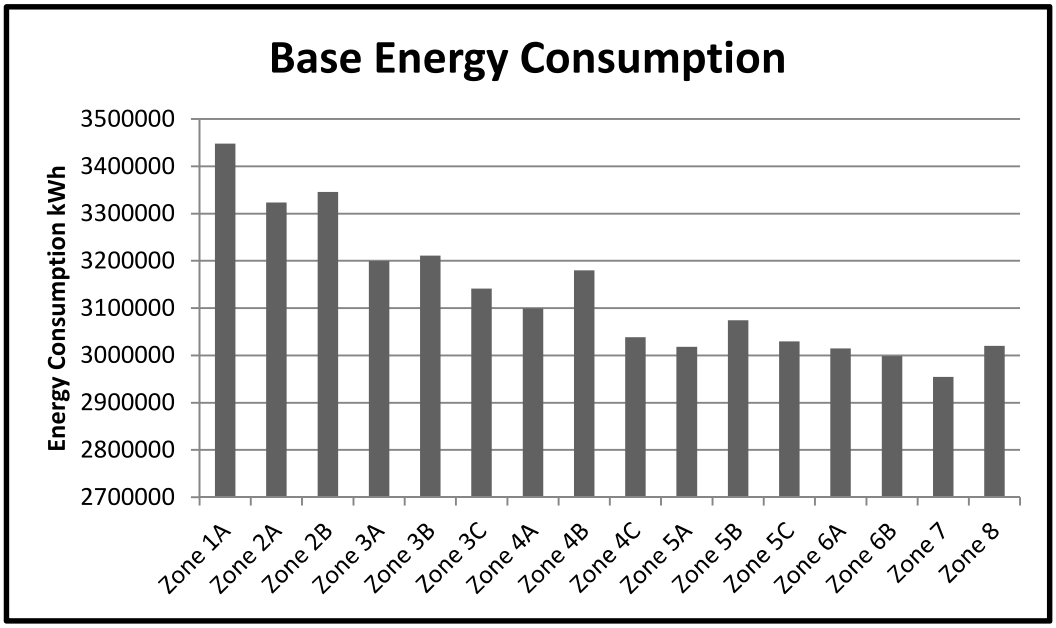

Figure 4.1 Base annual energy consumption of the building, in kWh, for each climate zone with no air-side economizer

Due to variations in the weather, a single airside economizer setpoint for every climate zone may or may not be optimal. To examine this the next group of simulations was the base case but now with an airside economizer activated with a fixed dry-bulb temperature control. However, eQUEST's internally-calculated high-limit setpoints differ by climate zone. Through an iterative process, the high limit was determined, set manually for each climate zone, and then remained constant throughout the iterations. The independent variable was then the low-limit; its value varied from the high limit temperature to an estimated low OA temperature for the otherwise repetitive simulations. Figures 4.2 through Figure 4.18 show the results of these building simulations where the economizer was now active.

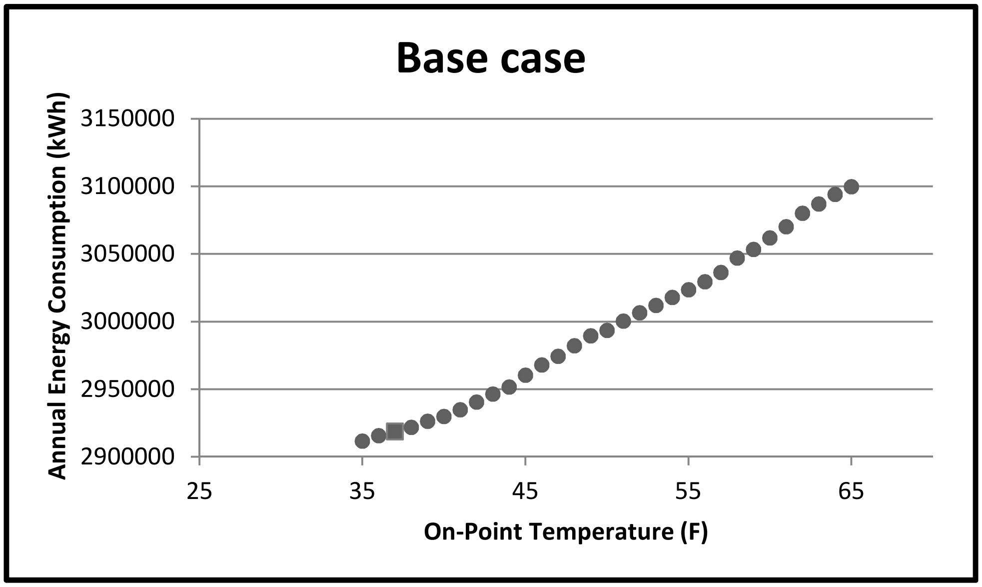

Figure 4.2 The base case's results, for Lenexa, KS, with the economizer now active, but with the low-limit varied

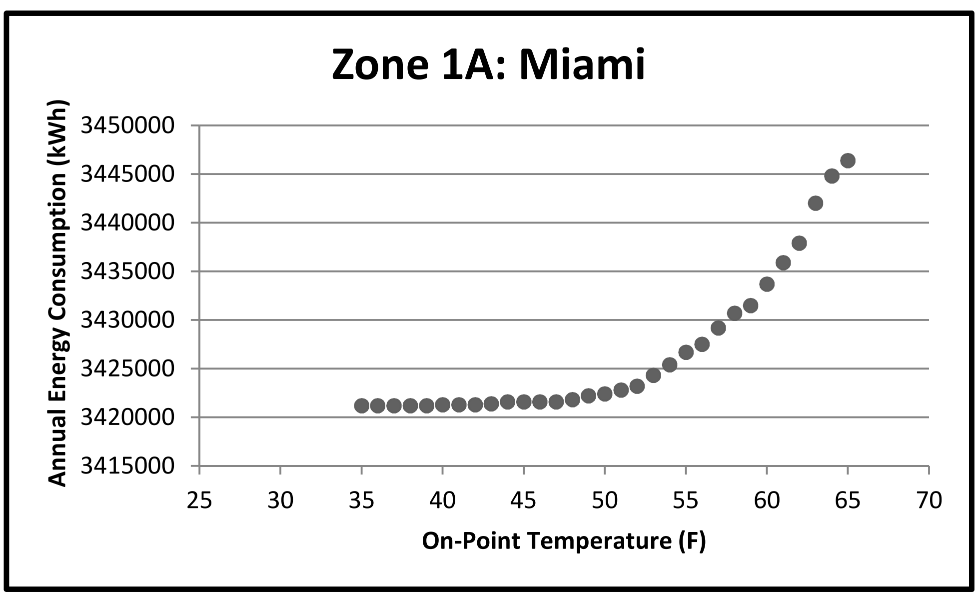

Figure 4.3 Zone 1A – Miami, FL

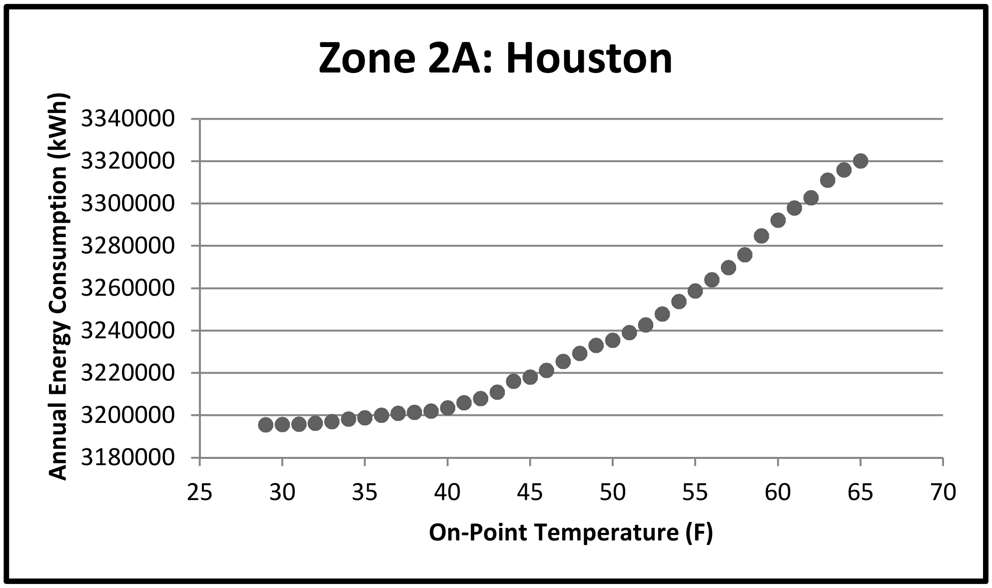

Figure 4.4 Zone 2A – Houston, TX

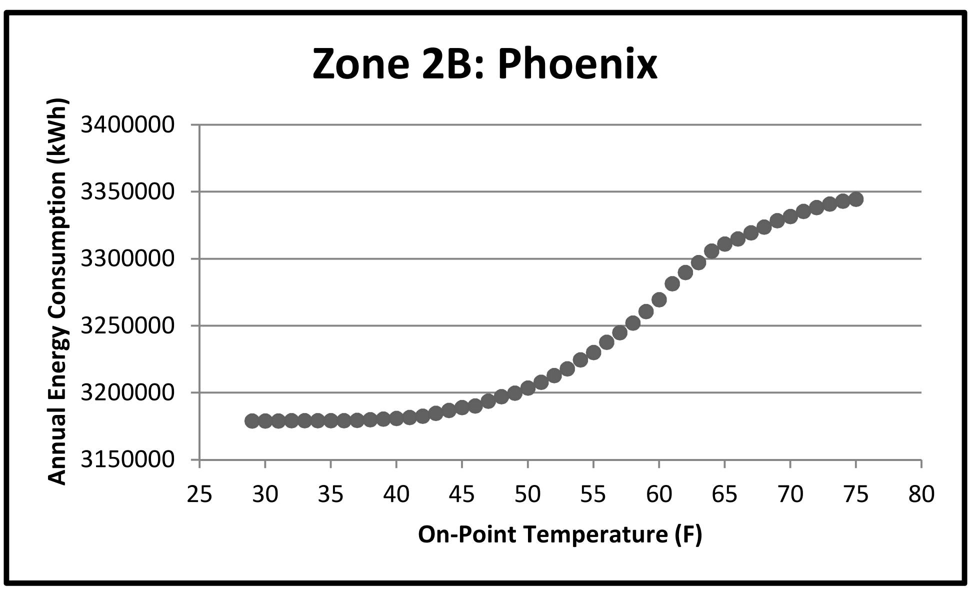

Figure 4.5 Zone 2B – Phoenix, AZ

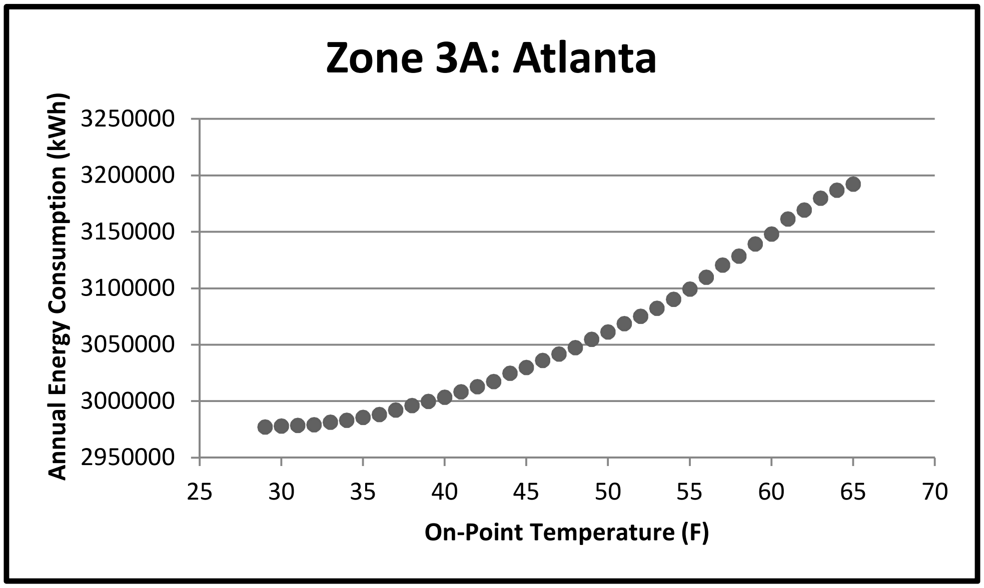

Figure 4.6 Zone 3A – Atlanta, GA

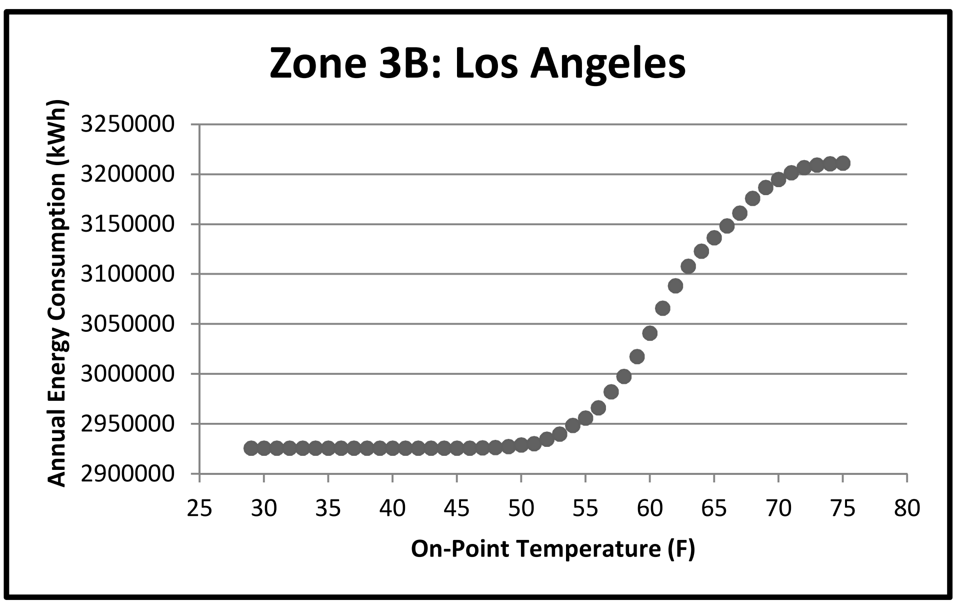

Figure 4.7 Zone 3B – Los Angeles, CA

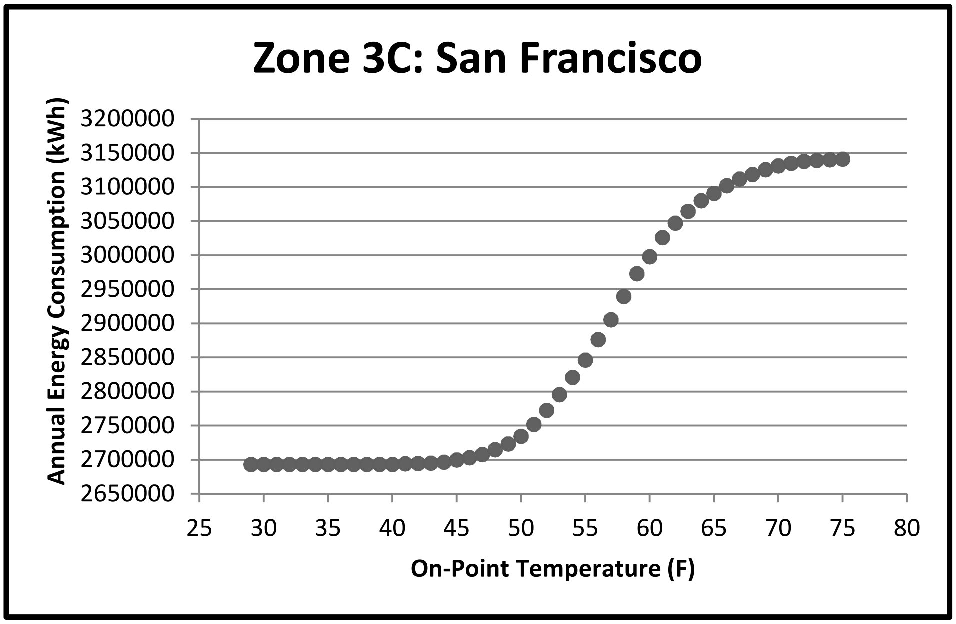

Figure 4.8 Zone 3C – San Francisco, CA

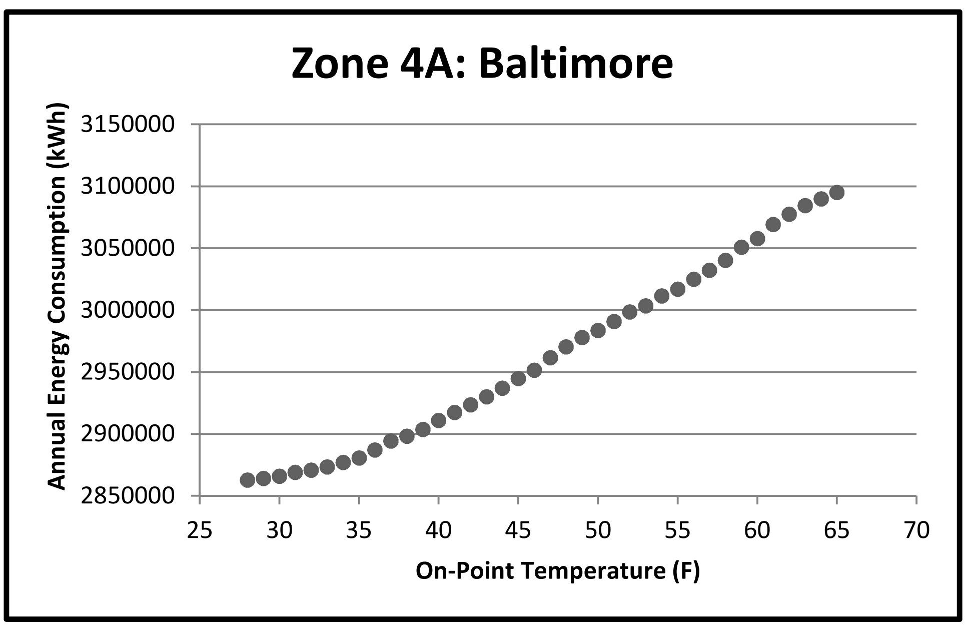

Figure 4.9 Zone 4A – Baltimore, MD

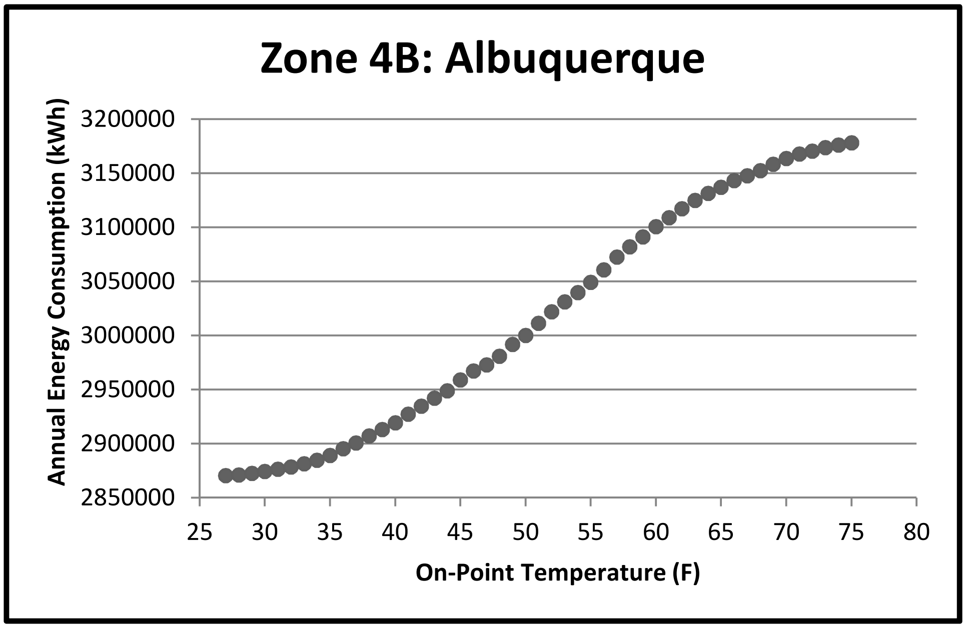

Figure 4.10 Zone 4B – Albuquerque, NM

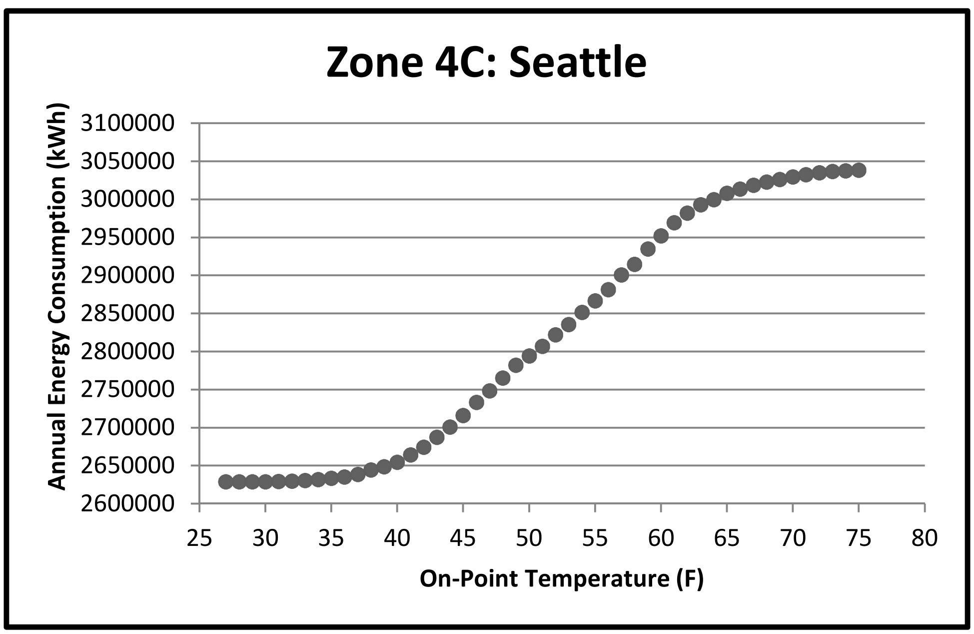

Figure 4.11 Zone 4C – Seattle, WA

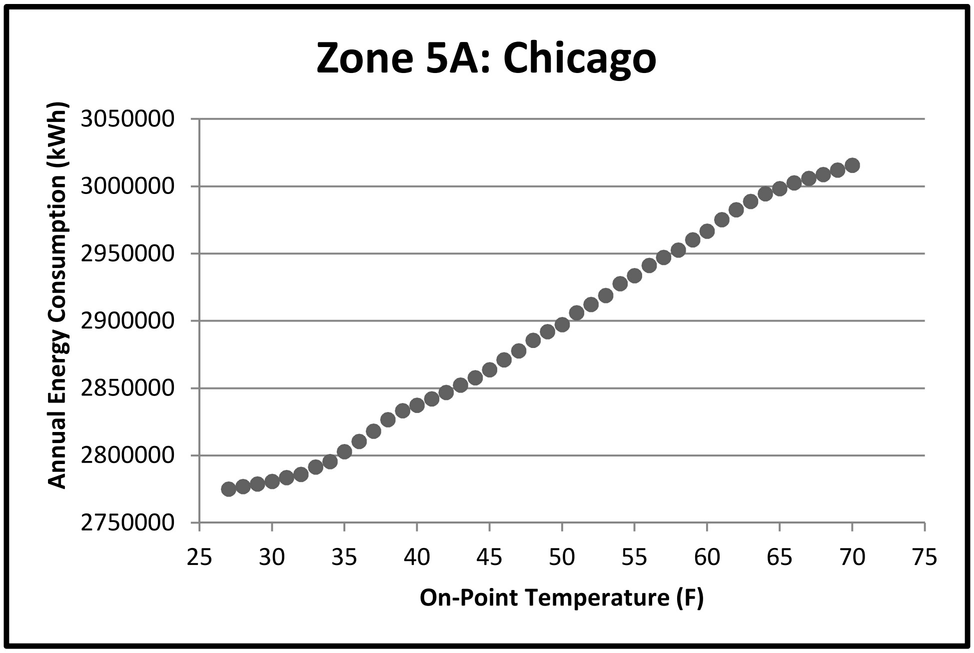

Figure 4.12 Zone 5A – Chicago, IL

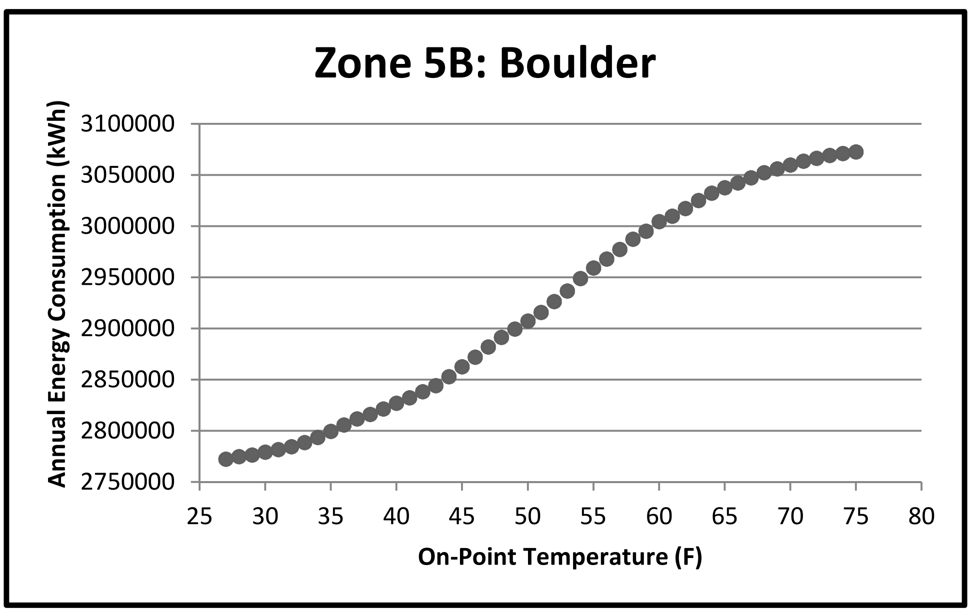

Figure 4.13 Zone 5B – Boulder, CO

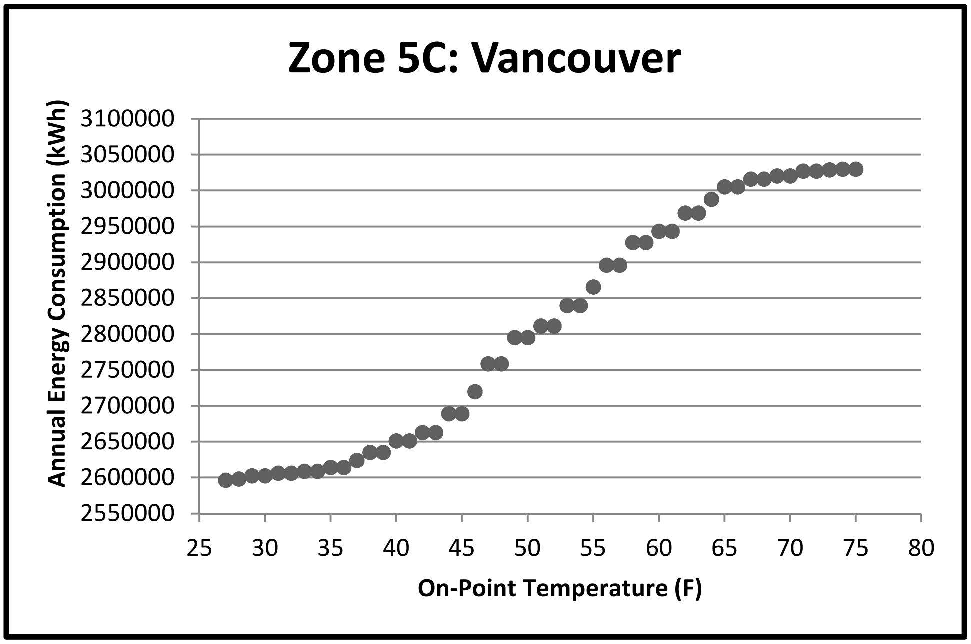

Figure 4.14 Zone 5C – Vancouver, BC

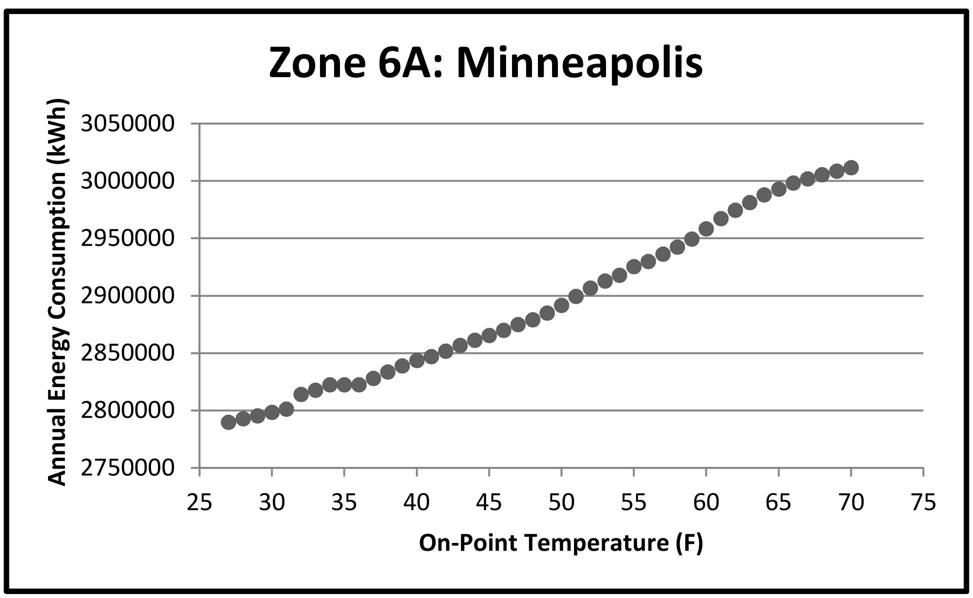

Figure 4.15 Zone 6A – Minneapolis, MN

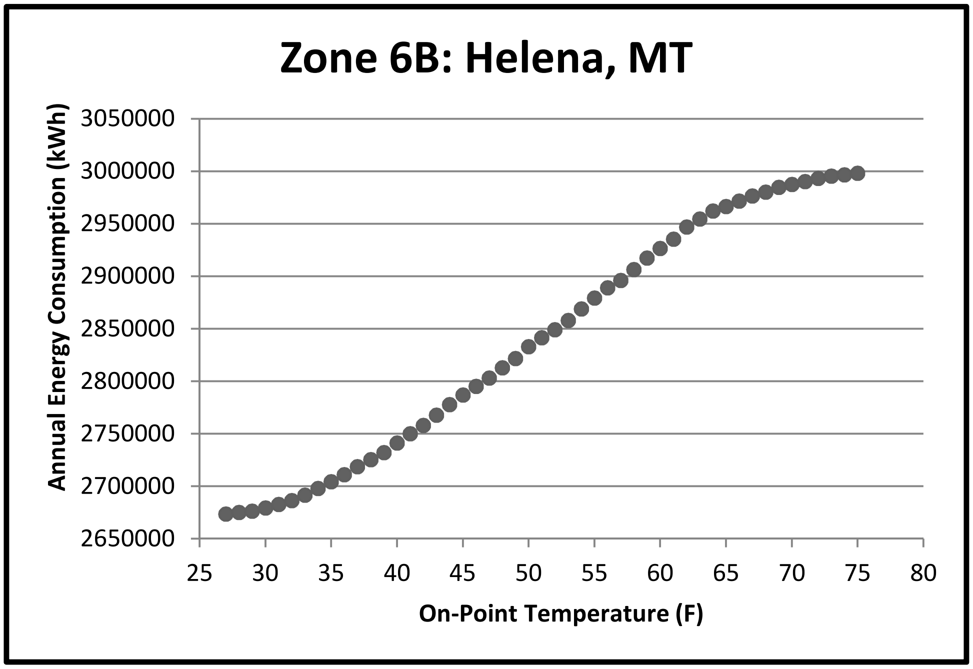

Figure 4.16 Zone 6B – Helena, MT

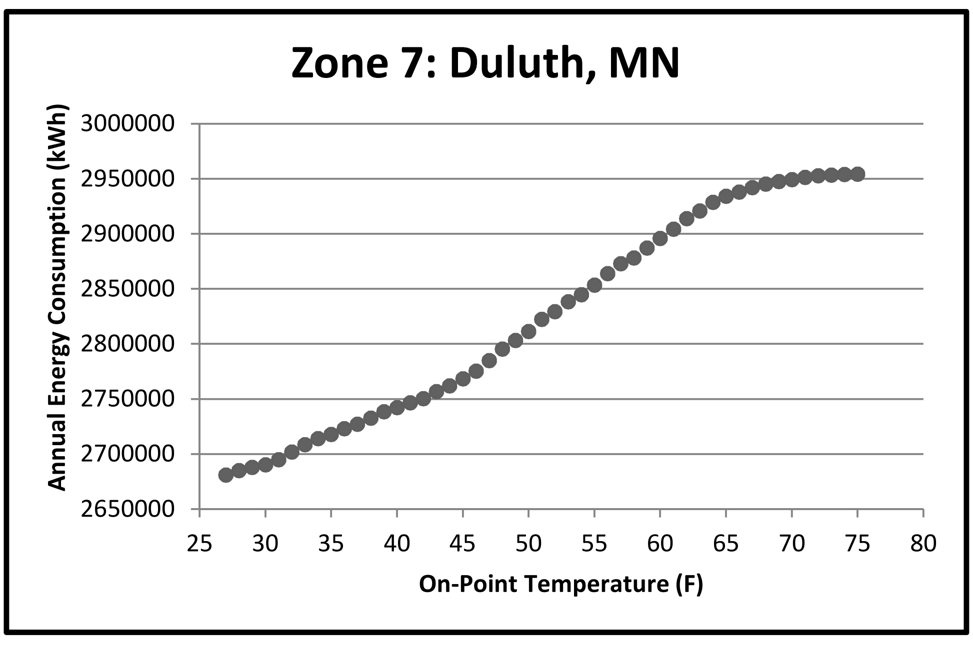

Figure 4.17 Zone 7 – Duluth, MN

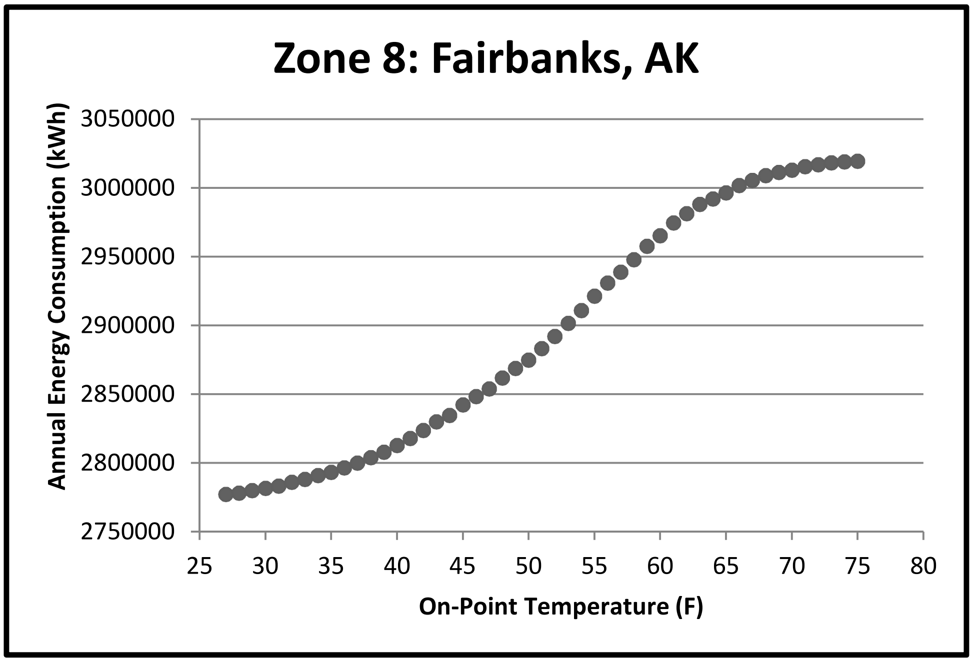

Figure 4.18 Zone 8 – Fairbanks, AK

The curves produced have several shapes. In Figure 4.8 for San Francisco, CA, the curve of the results were "S" shaped. But, in Figure 4.12 for Chicago, IL, the results are more linear. On the curves where the slope goes to near zero, in the left of each figure for lower OA temperatures, this implies the possible optimal low-limit where the slope starts to increase. But on other curves, seemingly for the colder climates, a minima is not explicit and those cases must still be further investigated. Ultimately, the goal of this project is to determine a practical way to estimate the optimal low limit. Calculating the balance point temperatures for the various locations, and then comparing those results to the simulations' findings is the next step in characterizing the low limit.

CHAPTER V

BALANCE POINT TEMPERATURE

According to ASHRAE Technical Committee (TC) 1.6, the balance point temperature (T_{BP}, or BPT; in °F or °C) is defined as “The outdoor [dry bulb] temperature at which a building’s heat loss to the environment is equal to internal heat gains from people, lights, and equipment… Internal-load dominated structures, like office buildings, may have balance points so low that the climate never overcomes their internal heat gain” (ASHRAE). The latter half of this statement implies for some buildings in certain climates they never require heating due to their high internal heat gains. The low-limit for an air-side economizer is when the OA air temperature alone allows comfortable conditions within the building to be maintained without the use of any additional mechanical heating or cooling (Utzinger and Wasley 1997), so the balance point temperature, a calculated or observed value, seems similar to the low-limit temperature of a drybulb economizer. But the balance point temperature is not measurable directly; it must be obtained through years of observation of a building. Or it may be predicted through calculations using multiple variables that correlate to a building’s design. Many thermal driving forces exist in a building, including heat transferred by radiation from the sun, heat released by occupancy, heat generated by lights and equipment, and conductive transfer of energy across the building enclosure. These energy flows can be evaluated under quasi-steady conditions, but a transient analysis that also accounts for heat storage and release from the masses within the building would better characterize the actual buildings’ performance in that particular climate.

The balance point temperature thus represents a practical energy balance of all of the aforementioned variables except the "thermal masses." Mathematically the balance point temperature, T_{BP}, is a combination of these variables (Utzinger et. al Wasley),

$$\boldsymbol{T_{BP} = T_{STAT} - T_{tempdiff}} \tag{5.1}$$

where T_{STAT} is the thermostat's setpoint temperature and $T_{tempdiff}$ is calculated from

$$\boldsymbol{T_{tempdiff} = \frac{Q_{IHG} + Q_{SOL}}{\widehat{U}_{BLDG}}} \quad . \tag{5.2}$$

T_{STAT} and $T_{tempdiff}$ have units of °F or °C. In Equation 5.2 Q_{IHG}, defined per unit floor area, is the buildings internal heat gain rate due to occupancy. Q_{SOL} is the rate of solar heat gain per unit floor area. $\widehat{U}_{BLDG}$ is the overall heat transfer across the building enclosure per unit floor area and per degree temperature difference. These three variables are constantly changing as they are affected by occupancy, time of day and year, weather conditions, and air exchange rates; $\widehat{U}_{BLDG}$ changes based on the current infiltration and ventilation rate. Because these variables change, the balance point temperature does vary somewhat with time for a particular building and climate. When only one balance point temperature is stated, which is the norm, it represents a compilation or possibly only one set of conditions such as those for the "worst case." So the method for determining the low-limit presented here should be considered as a refined starting point, and the value should be adjusted after observing the operation of the particular HVAC system over time and under various conditions.

The internal heat gains can be separated into their sensible and latent components.

Humans emit moisture at varying rates due to their different levels of activity. Typically, one of two methods is used to determine the heat gains of occupancy. The first method is a lookup-table in the Fundamentals volume of the ASHRAE Handbook (ASHRAE 2013). The chapter “Nonresidential Cooling and Heating Load Calculations” provides a table that gives rates at which heat and moisture are emitted based on a human’s different level of activity. The second, more fundamental method is given in ASHRAE Standard 55-2013 and uses metabolic rates. Metabolic rate, as defined in ASHRAE Standard 55-2013, is “the rate of transformation of chemical energy into heat and mechanical work by metabolic activities within an organism, usually expressed in terms of unit area of the total body surface” (ASHRAE Standard 55-2013).

Another part of the buildings’ internal heat gains is from lights and equipment. For lights, the electrical energy consumed by each fixture is ultimately equivalent to the rate of heat dissipated; some of the heat gain becomes a load immediately while the remainder is stored in masses and then released later. Similarly for equipment, the heat released is equal to the energy each piece of equipment consumes, but with certain equipment, such as steam tables and coffee pots, there’s also a latent component; the building’s masses can store and release moisture as well as the sensible heat. As previously noted, Q_{IHG} is the total internal heat gain per unit floor area, and is

$$\boldsymbol{Q_{IHG} = Q_{PEOPLE} + Q_{LIGHTS} + Q_{EQUIP}} \tag{5.3}$$

where Q_{PEOPLE} is the heat gain from people occupying the building per unit floor area, Q_{LIGHTS} is the heat gain from the lights used during occupancy per unit floor area, and Q_{EQUIP} is the heat gain from equipment per unit floor area. These values can be characterized as steady-state or transient. The DOE-2 based program eQUEST uses the “quasi-steady” approach – hourly-averaged values are found, and energy storage and release is characterized with relatively simple time-lag factors that vary with the construction materials used.

The next variable in the balance point temperature equation is the solar heat gain, Q_{SOLAR} that can also be described by

$$\boldsymbol{Q_{SOL} = q_{sol}/A_{floor}} \quad . \tag{5.4}$$

Q_{SOL} is, of course, transient. But for the ultimate purpose of this study, a simplified way to characterize or gather the solar heat gain was needed. Such will be presented later in this thesis.

The overall building heat transfer coefficient has multiple components too (Utzinger et. al 2011 Wasley):

$$\boldsymbol{\widehat{U}_{bldg} = \widehat{U}_{wall} + \widehat{U}_{roof} + \widehat{U}_{glzg} + \widehat{U}_{grnd} + \widehat{U}_{vent}} \tag{5.5}$$

where $\widehat{U}_{wall}$ represents the floor area-averaged heat transfer coefficient through the buildings walls. It is determined from

$$\boldsymbol{\widehat{U}_{wall} = (U_{wall} * A_{wall})/A_{floor}} \tag{5.6}$$

where U_{wall} is the overall heat transfer coefficient for the exterior walls of the building. Second, A_{wall} is the total wall surface area and A_{floor} is the overall building floor area. This is the overall U-value for the entire building; the equations are not normally applied to floors individually. The other components of equation 5.5 are $\widehat{U}_{roof}$, $\widehat{U}_{glzg}$, and $\widehat{U}_{grnd}$; determining all three follow the same concept of the building heat transfer rate through the wall. The equations for these three variables are thus:

$$\widehat{U}_{roof} = (U_{roof} * A_{roof})/A_{floor} \quad \textbf{(5.7)}$$

$$\widehat{U}_{glzg} = (U_{glzg} * A_{glzg})/A_{floor} \quad \textbf{(5.8)}$$

$$\widehat{U}_{grnd} = (U_{grnd} * A_{grnd})/A_{floor} \quad \textbf{(5.9)}$$

All of the values required to perform the balance point temperature calculation can fortunately be extracted directly from Trace^{TM}700's load calculation results, and an HVAC system designer would already be performing this load analysis using Trace^{TM} or a similar tool. Doing such for this study's building, the balance point temperature was calculated for the base case using a worksheet that was created to organize and solve the equations. For adoption by designers it was important to create a simple yet effective way to perform the needed calculations. The spreadsheet appears in Figure 5.1 and uses the inch-pound (I-P) units shown in Table 5.1.

Table 5.1 Building Balance Point Temperature Worksheet Units

Variable	Units
$\boldsymbol{U_{wall}}$	Btu/Hr/sf^2/°F
$\boldsymbol{A_{wall}}$	sf^2
$\boldsymbol{A_{floor}}$	sf^2
$\boldsymbol{U_{roof}}$	Btu/Hr/ sf^2/°F
$\boldsymbol{A_{roof}}$	sf^2
$\boldsymbol{U_{glzg}}$	Btu/Hr/ sf^2/°F
$\boldsymbol{A_{glzg}}$	sf^2
$\boldsymbol{U_{grnd}}$	Btu/Hr/LF/°F
$\boldsymbol{A_{grnd}}$	sf^2
$\boldsymbol{\widehat{U}_{wall}}$	Btu/Hr/°F/ sf^2
$\boldsymbol{\widehat{U}_{roof}}$	Btu/Hr/°F/ sf^2
$\boldsymbol{\widehat{U}_{glzg}}$	Btu/Hr/°F/ sf^2
$\boldsymbol{\widehat{U}_{grnd}}$	Btu/Hr/°F/ sf^2
$\boldsymbol{\widehat{U}_{bldg}}$	Btu/Hr/°F/ sf^2
$\boldsymbol{q_{people}}$	Btu/Hr
$\boldsymbol{q_{light}}$	Btu/Hr
$\boldsymbol{q_{equip}}$	Btu/Hr
$\boldsymbol{Q_{PEOPLE}}$	Btu/Hr/ sf^2
$\boldsymbol{Q_{LIGHTS}}$	Btu/Hr/ sf^2
$\boldsymbol{Q_{EQUIP}}$	Btu/Hr/ sf^2
$\boldsymbol{Q_{IHG}}$	Btu/Hr/ sf^2
$\boldsymbol{q_{sol}}$	Btu/Hr
$\boldsymbol{Q_{SOL}}$	Btu/Hr/ sf^2
$\boldsymbol{T_{STAT}}$	°F
$\boldsymbol{T_{tempdiff}}$	°F
$\boldsymbol{T_{BP}}$	°F

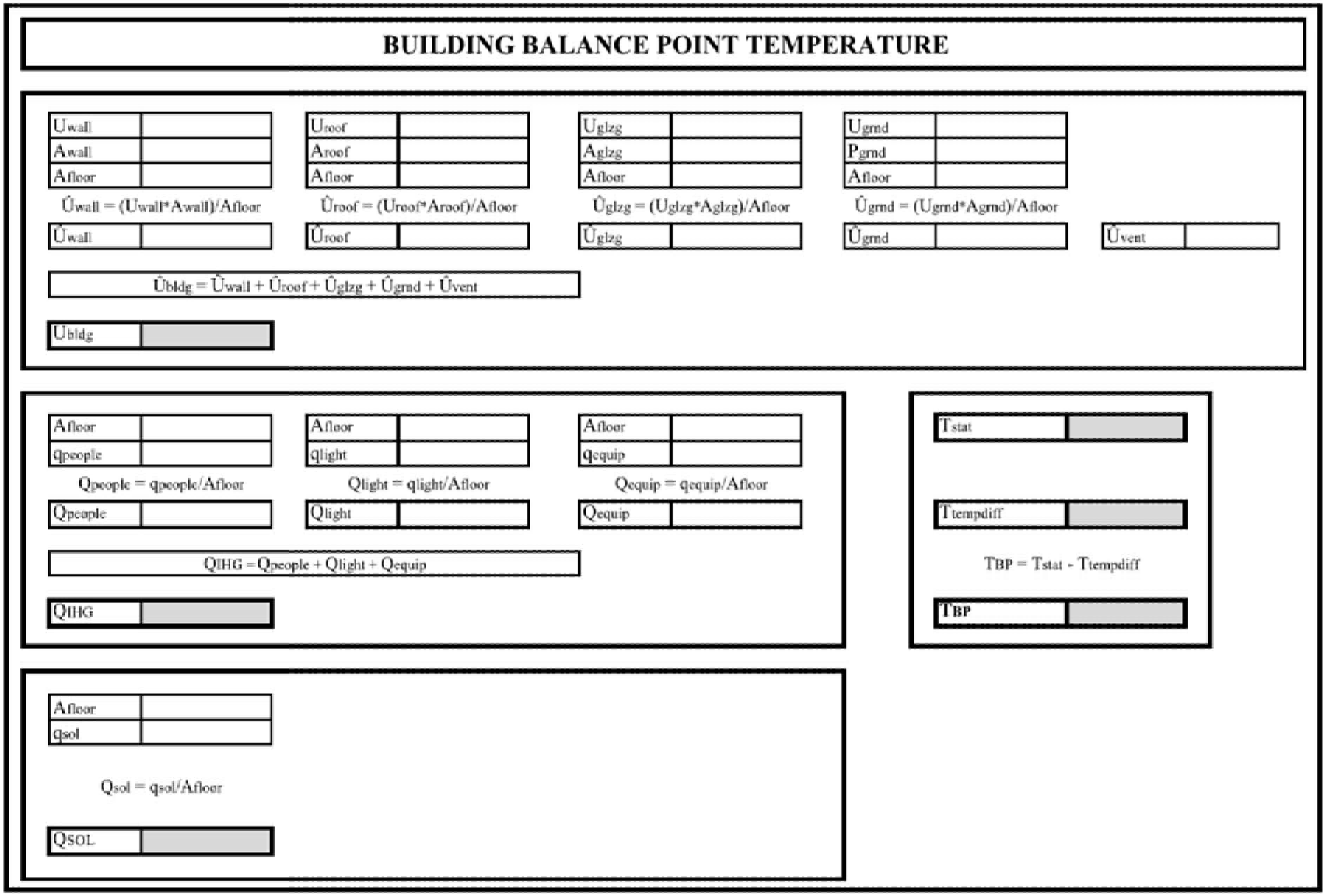

Figure 5.1: Building Balance Point Temperature Worksheet

When using Trace^{TM}700, its resulting “System Checksums” report contains the majority of the data required to calculate the building’s balance point temperature. An example, using this study’s base case, follows.

The first value needed from the System Checksums report is the floor square footage, A_{floor}. Figure 5.2 shows where to find this value on the report. The needed U-values were determined based on the construction of the building.

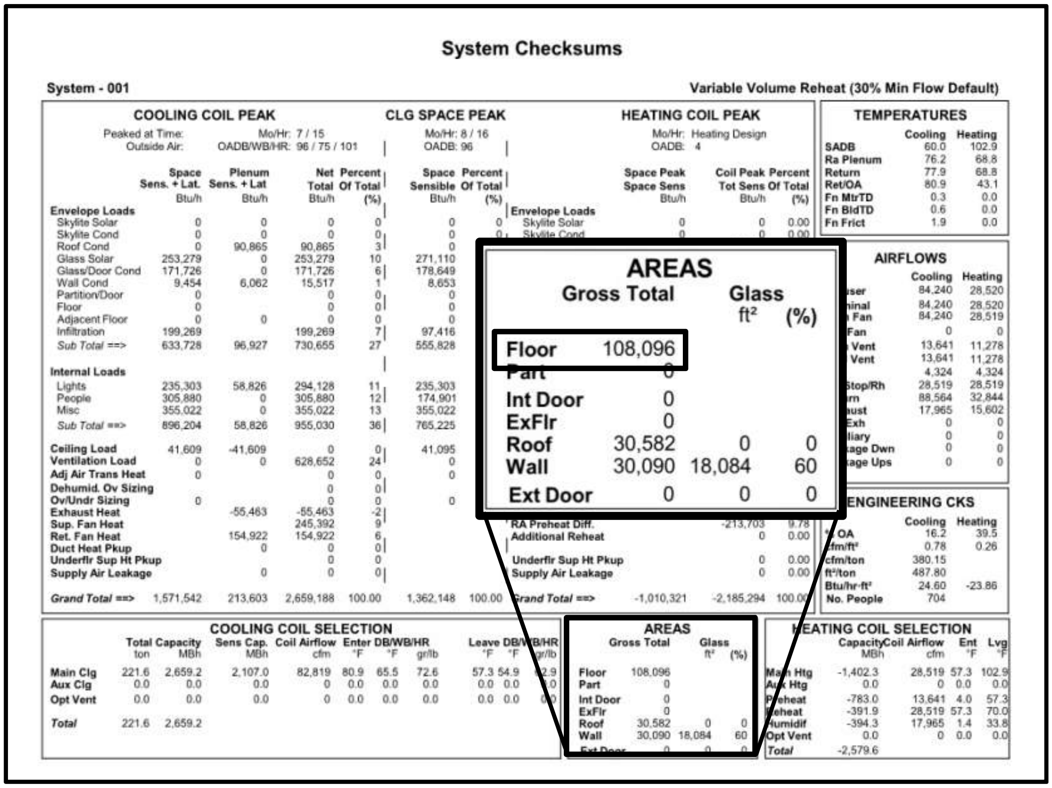

Figure 5.2 Collecting floor area square footage from a Trane Trace® System Checksums report

Figure 5.3 indicates where to enter the U-Values and the floor area, on the template.

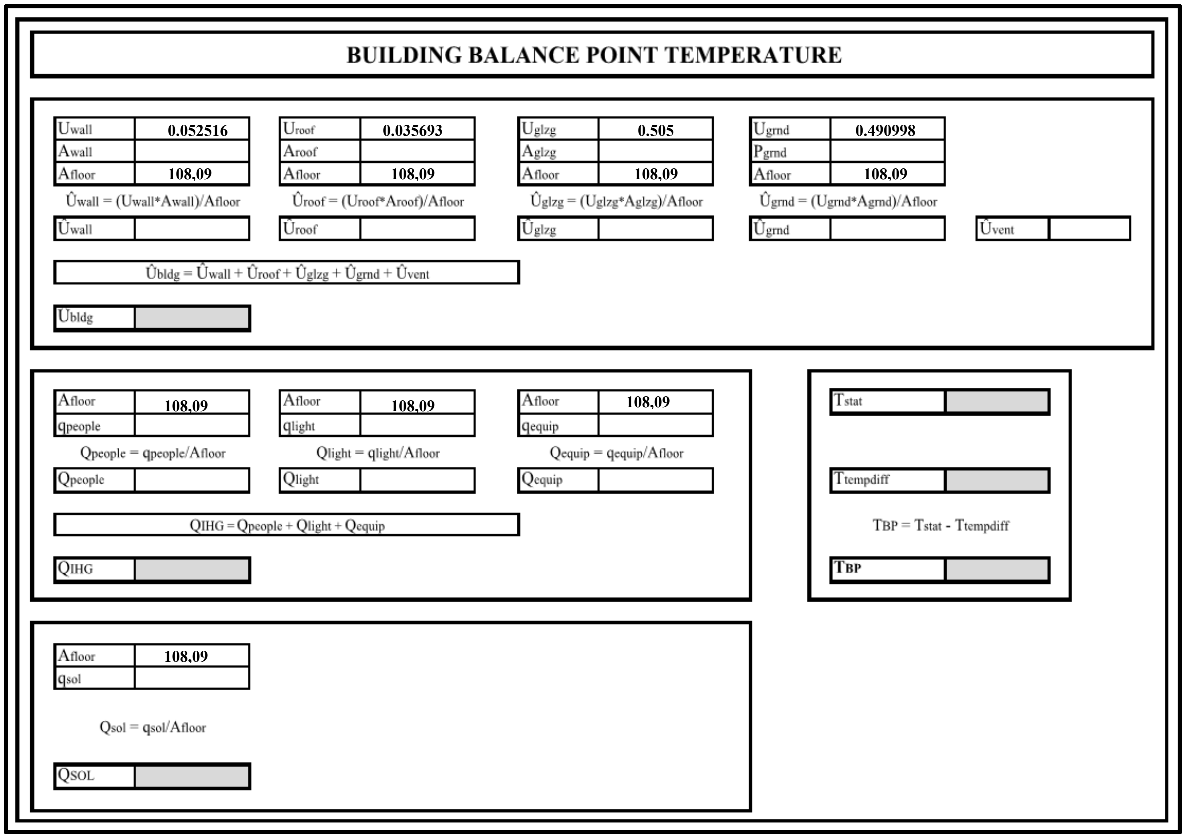

Figure 5.3 Building balance point temperature worksheet with the example's U-values and floor area entered

The next step is to gather the total wall area, roof area, and glazing area from the System Checksums report. This information can be found on the System Checksums report as seen in Figure 5.4. Figure 5.5 shows the values filled in on the worksheet.

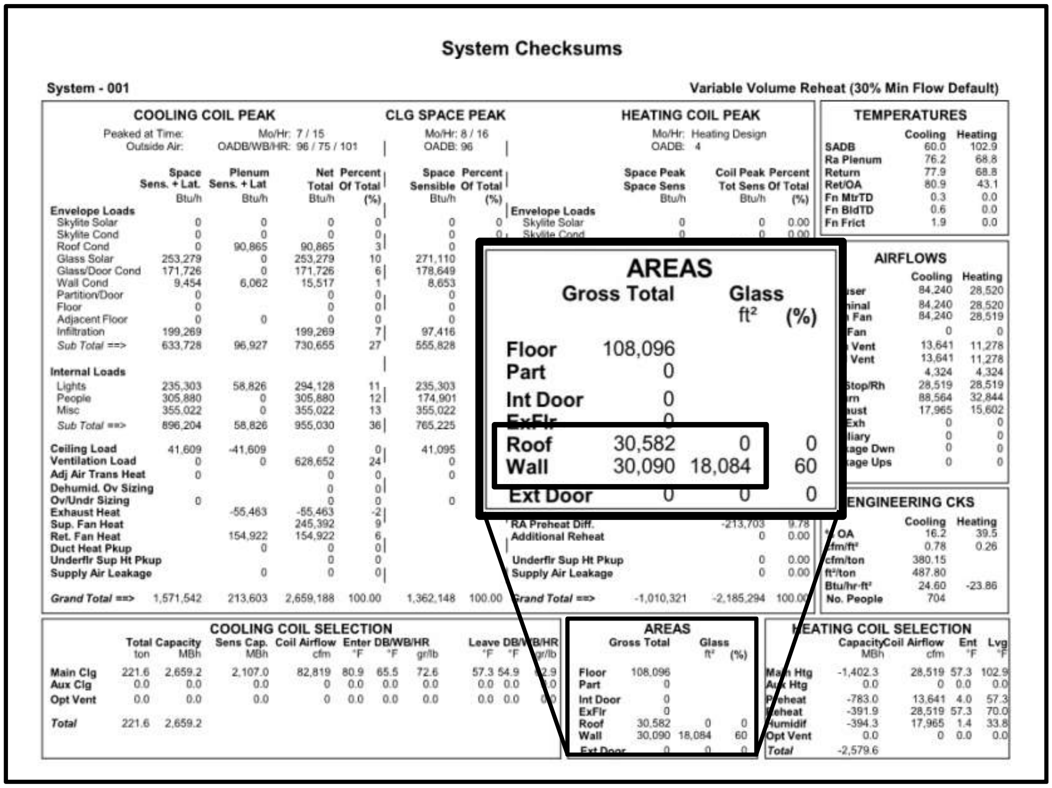

Figure 5.4 Collecting wall, roof, and glazing areas from the System Checksums report.

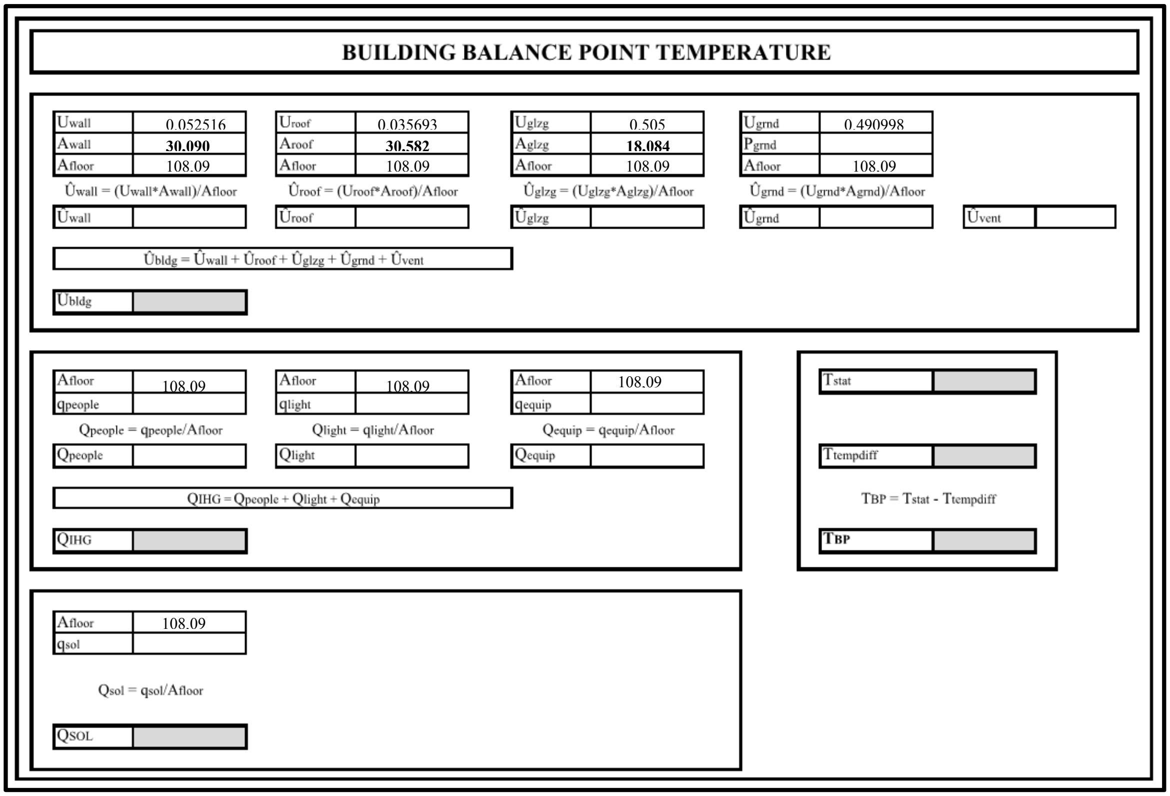

Figure 5.5 Building balance point temperature worksheet with the wall, roof, and glazing areas added

While almost all needed values can be found from earlier input data or the System Checksums report, a couple values cannot. Those two values are the perimeter of the building and the infiltration and ventilation heat transfer rate. The perimeter of the building is a simple calculation using the exterior dimensions obtained from the architectural plans. The building's air exchange heat transfer rate can be challenging to determine. For the ultimate practical result of this project – a design method for finding the low-limit -- a conservative estimate was used. The value chosen for this building's air exchange heat transfer rate was 0.13 Btu/hr/ft^2/°F.

With these values determined, they can then be added to the Building Balance Point Temperature Worksheet as shown in Figure 5.6.

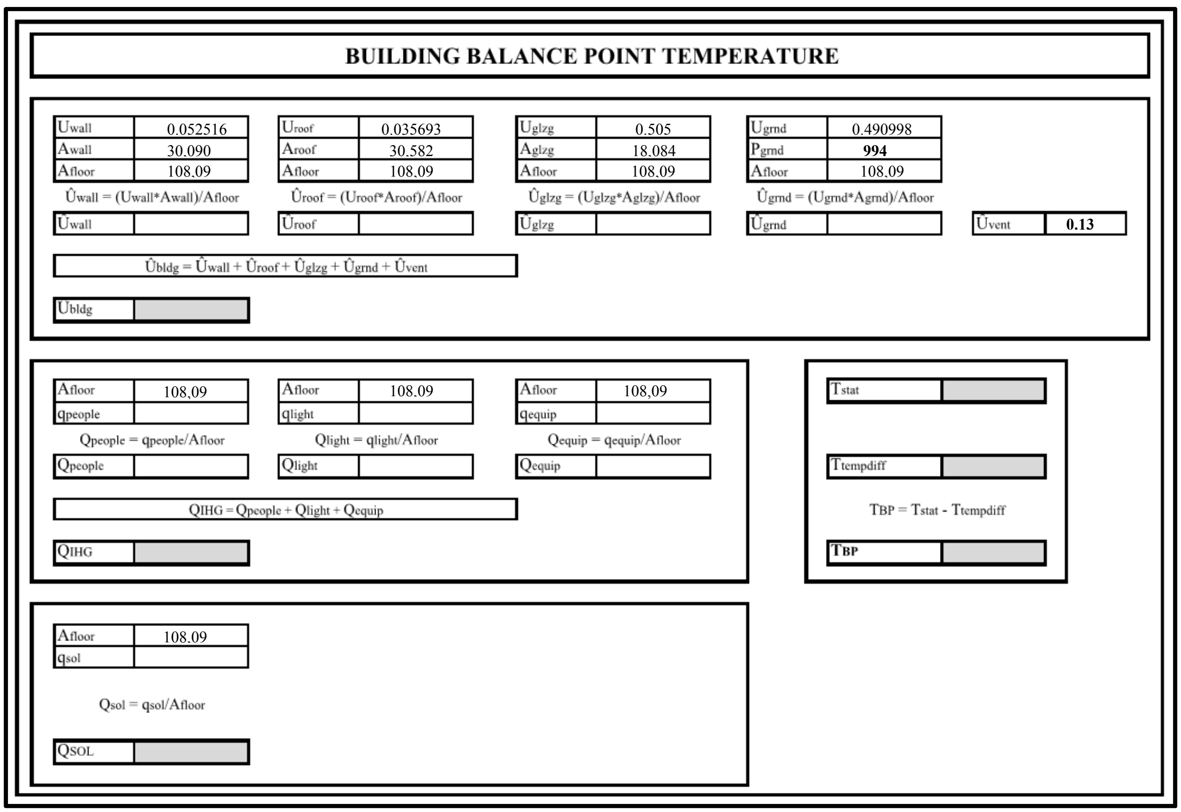

Figure 5.6 Building balance point temperature worksheet with the building's perimeter and air exchange heat transfer coefficient added

The next step is to collect the internal heat gain rates. Again, the internal heat gain rates for the building are those from people, lights, and equipment. Figure 5.7 shows where those values are found on the System Checksums report of Trane Trace®.

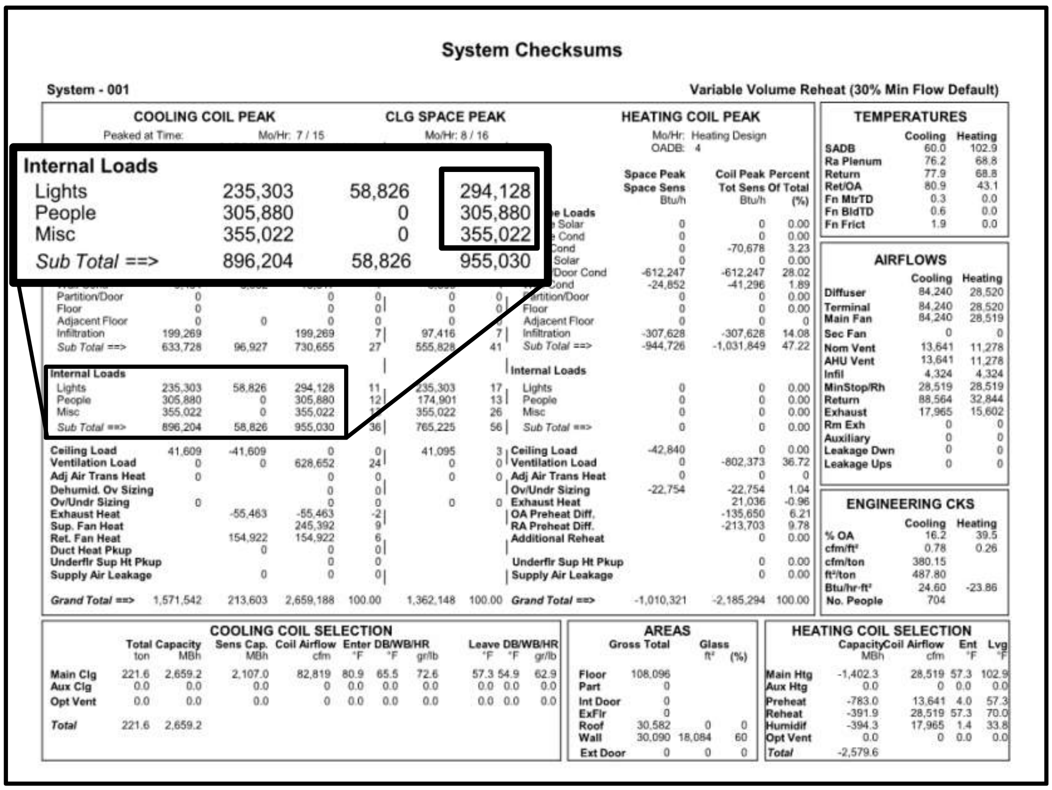

Figure 5.7 Collecting internal heat gain rates from the System Checksums report

The last value needed for the worksheet is the solar heat gain rate. As with most of the other data values, the solar heat gain rate can fortunately be found on the System Checksums report. Figure 5.8 gives the location of the solar heat gain rate on the report.

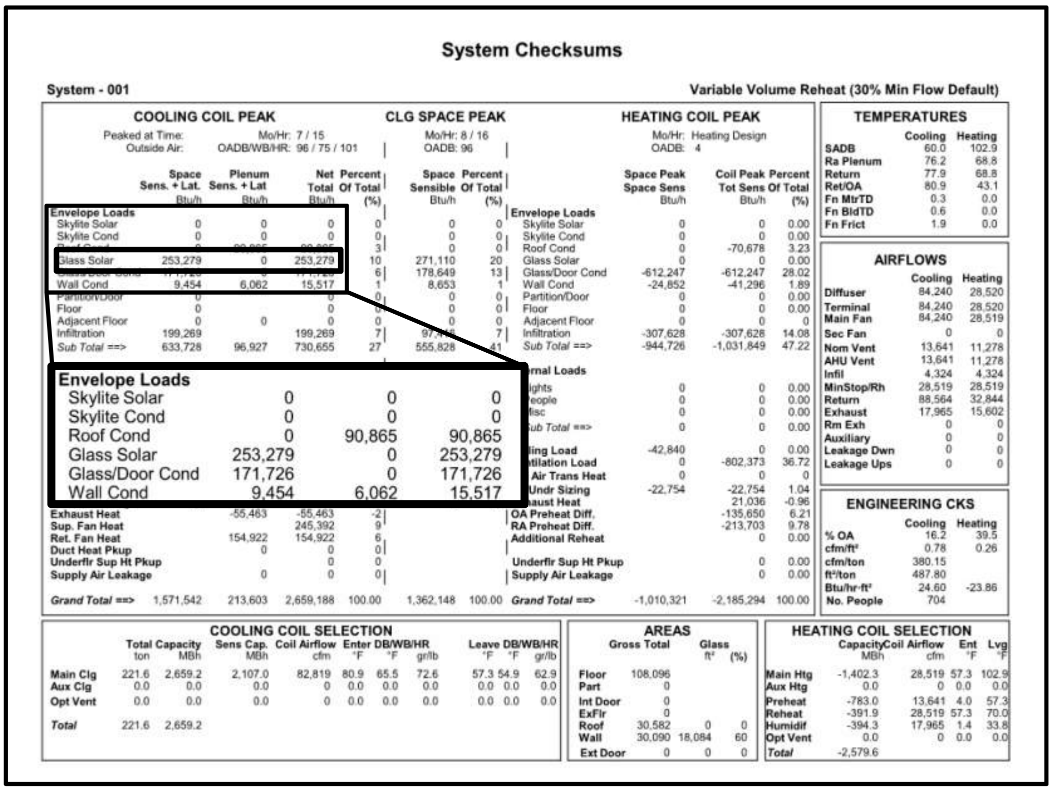

Figure 5.8 Collecting the solar heat gain rate from the System Checksums report

The Building Balance Point Temperature Worksheet, now complete for the base case of this study, appears in Figure 5.9.

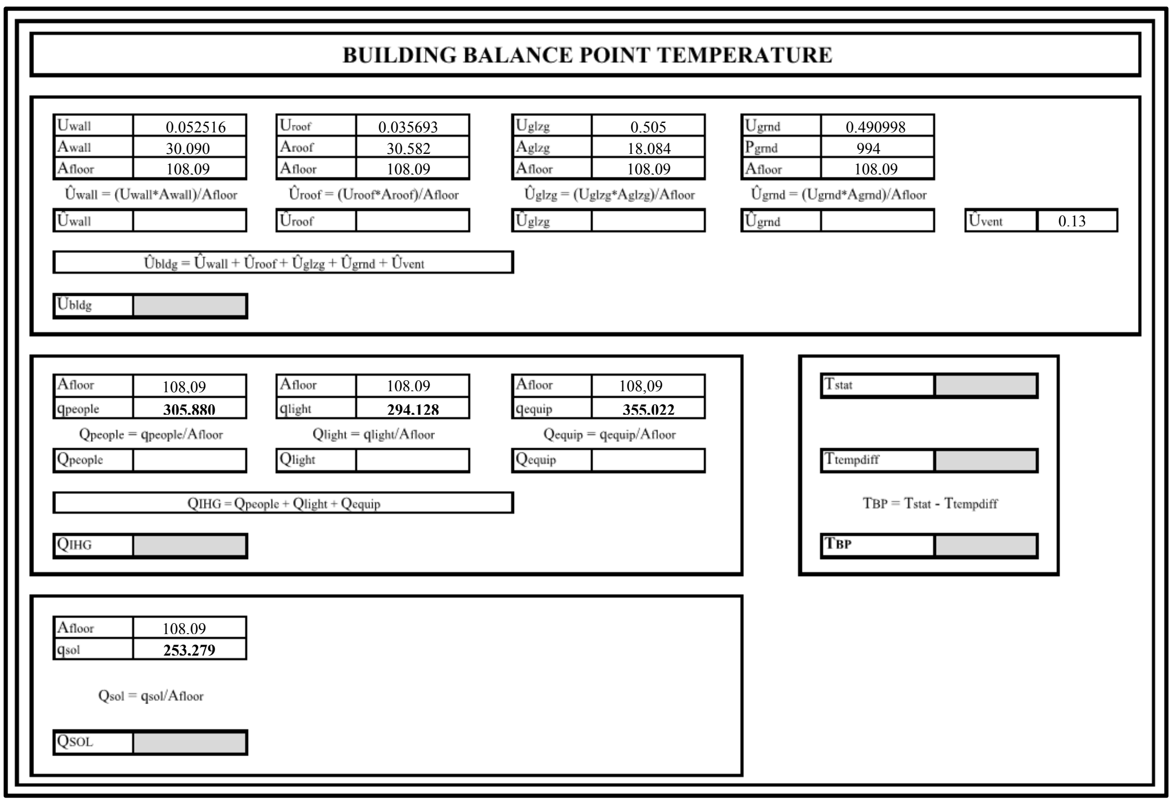

Figure 5.9 Building balance point temperature worksheet with the internal and solar heat gains added

The next step, via cell formulas in the spreadsheet, is the calculation using all the data collected. The building heat transfer rate calculation is separated into heat transfer through walls, roof, glazing, ground, and air exchange. The normalized heat transfer rate through the building walls is:

$$\widehat{U}_{wall} = (U_{wall} * A_{wall})/A_{floor} \quad \textbf{(5.10)}$$

Figure 5.10 gives the results for this example.

Uwall	0.052516	Btu/Hr/ft²/°F
Awall	30,090	ft²
Afloor	108,096	ft²

Ûwall = (Uwall*Awall)/Afloor

Ûwall	0.014619	Btu/Hr/°F/ ft²

Figure 5.10 Building balance point temperature worksheet's wall heat rate calculation for the base case

The roof's heat transfer rate is:

$$\widehat{U}_{roof} = (U_{roof} * A_{roof})/A_{floor} \quad \textbf{(5.11)}$$

Figure 5.11 shows the results for the base case's roof.

U_{roof}	0.035693	Btu/Hr/ft^2/°F
A_{roof}	30,582	ft^2
A_{floor}	108,096	ft^2

$\hat{U}_{roof} = (U_{roof} * A_{roof})/A_{floor}$

$\hat{U}_{roof}$	0.010098	Btu/Hr/°F/ ft^2

Figure 5.11 Building balance point temperature worksheet roof heat rate calculation for the base case

The next two parts of the worksheet pertain to the glazing and the ground or slab:

$$\hat{U}_{glzg} = (U_{glzg} * A_{glzg})/A_{floor} \quad \textbf{(5.12)}$$

$$\hat{U}_{grnd} = (U_{grnd} * A_{grnd})/A_{floor} \quad \textbf{(5.13)}$$

Similar to that for the walls and the roof Figures 5.12 and 5.13 show the calculations for glazings and the ground contact.

U_{glzg}	0.505	Btu/Hr/ft^2/°F
A_{glzg}	18,084	ft^2
A_{floor}	108,096	ft^2

$$\hat{U}_{glzg} = (U_{glzg} * A_{glzg}) / A_{floor}$$

$\hat{U}_{glzg}$	0.84484	Btu/Hr/°F/ ft^2

Figure 5.12 Building balance point temperature worksheet's glazing heat rate calculation for the base case

U_{grnd}	0.490998	Btu/Hr/ft/°F
P_{grnd}	994	ft
A_{floor}	108,096	ft^2

$$\hat{U}_{grnd} = (U_{grnd} * A_{grnd}) / A_{floor}$$

$\hat{U}_{grnd}$	0.004515	Btu/Hr/°F/ ft^2

Figure 5.13 Building balance point temperature worksheet's ground heat rate calculation for the base case

Finally, the last needed value is the normalized ventilation heat transfer rate, and it is already in a directly useable form, $\widehat{U}_{vent} = 0.13$.

With the needed data calculated, the overall building heat transfer rate is:

$$\widehat{\boldsymbol{U}}_{bldg} = \widehat{\boldsymbol{U}}_{wall} + \widehat{\boldsymbol{U}}_{roof} + \widehat{\boldsymbol{U}}_{glzg} + \widehat{\boldsymbol{U}}_{grnd} + \widehat{\boldsymbol{U}}_{vent} \quad \textbf{(5.14)}$$

The total for $\widehat{U}_{bldg}$ for the base case is 0.243716 BTU/h·ft^2·°F.

The next portion of the Building Balance Point Temperature calculation produces the overall internal heat gains for the building. The first component of this value is the internal heat gain due to occupancy:

$$\boldsymbol{Q}_{people} = \boldsymbol{q}_{people}/\boldsymbol{A}_{floor} \quad \textbf{(5.15)}$$

Similarly, the internal heat gains due to lights and equipment are needed:

$$\boldsymbol{Q}_{lights} = \boldsymbol{q}_{lights}/\boldsymbol{A}_{floor} \quad \textbf{(5.16)}$$

$$\boldsymbol{Q}_{equip} = \boldsymbol{q}_{equip}/\boldsymbol{A}_{floor} \quad \textbf{(5.17)}$$

With these data, the internal heat gains are determined within the worksheet, as shown in Figures 5.14 through 5.16.

A_{floor}	108,096	ft^2
q_{people}	305,880	Btu/Hr

$$Q_{people} = q_{people}/A_{floor}$$

Q_{people}	2.83	Btu/Hr/ ft^2

Figure 5.14 Building balance point temperature worksheet's occupancy heat transfer rate, calculation for the base case

A_{floor}	108,096	ft^2
q_{light}	294,128	Btu/Hr

$$Q_{light} = q_{light}/A_{floor}$$

Q_{light}	2.72	Btu/Hr/ ft^2

Figure 5.15 Building balance point temperature worksheet's lighting heat transfer rate calculation for the base case

Afloor	108,096	ft^2
qequip	355,022	Btu/Hr

Qequip = qequip/Afloor

Qequip	3.28	Btu/Hr/ ft^2

Figure 5.16 Building balance point temperature worksheet's equipment heat transfer rate calculation for the base case

After completion of the three internal heat gain calculations, the overall building internal heat gain is:

$$\boldsymbol{Q_{IHG} = Q_{people} + Q_{light} + Q_{equip}} \tag{5.18}$$

The calculations in the third section of the Building Balance Point Temperature Worksheet, for the solar energy, are similar to that for the internal heat gains. This quick calculation of the solar heat gains, that are dependent on the location of the building, is:

$$\boldsymbol{Q_{SOL} = q_{sol}/A_{floor}} \tag{5.19}$$

For the base case, the result of solving the equation is shown in Figure 5.17.

A_{floor}	108,096	ft^2
q_{sol}	253,279	Btu/Hr

$Q_{sol} = q_{sol}/A_{floor}$

Q_{SOL}	2.34	Btu/Hr/ ft^2

Figure 5.17 Building balance point temperature worksheet's solar heat transfer rate calculation for the base case

The thermostat's setpoint temperature for cooling mode used in the modeling software was next needed for the final section's calculations. With it, the temperature difference due to the internal and solar heat gains, $T_{E\text{-}Flow}$, is found:

$$\boldsymbol{T_{E-FLOW} = \frac{Q_{IHG} + Q_{SOL}}{\hat{U}_{BLDG}}} \qquad \textbf{(5.20)}$$

Once this value for the temperature has been calculated, the balance point temperature for the building and its particular geographic location can be determined:

$$\boldsymbol{T_{BP} = T_{STAT} - T_{E-FLOW}} \qquad \textbf{(5.21)}$$

The worksheet's, balance point temperature calculation, for the base case, is given in Figure 5.18.

$T_{tempdiff}$	45.9	°F

$$T_{BP} = T_{stat} - T_{tempdiff}$$

T_{BP}	29.1	°F

Figure 5.18 Balance point temperature worksheet's final calculation for the base case

This 29.1°F temperature represents the typical outdoor air temperature where the base case building, when located in Lenexa, KS, would changeover from heating to cooling mode. By comparison, modern single family homes in the Midwestern United States often have balance point temperatures of around 50°F. Office buildings typically have much higher internal heat gains so their balance point temperatures are much lower. This lower balance point allows them to benefit more from the use of air-side economizers because there are many more hours of the year when the outside air temperature is between the balance point and the design supply air temperature for cooling, typically 55°F.

Also note that, for buildings with high internal and solar loads, the balance point temperature can be below 32°, thus water-fill coils or humidifiers in their HVAC systems can freeze. To prevent such damage HVAC designers often specify preheat coils that activate when the OA or MA crossing them falls below 40°F or so. Such heating of the OA would negate some of the benefit of the economizer and increase energy consumption. So better would be to eliminate the possibility of freezing through effective mixing of the cold OA and the warm CA, or to fully eliminate the use of water downstream of the OA intake during cold conditions. Some steam heating coils are fully self-draining, for example.

CHAPTER VI

ECONOMIZER SET POINT ANALYSIS

An air-side economizers' high limit has typically been set to 68 to 70°F, with toward the former in humid climates and toward the latter in drier. The low-limit should vary with climate zone, however, due to the weather's significant effect on a building's cooling load as described previously. Recently, the selection of the high limit has become more refined as shown through ASHRAE Standard 90.1-2013 (ASHRAE 2013). Designers need similar guidance for the low-limit. Up to now no values for the low limit of the airside economizer have been suggested in the Standard. Part of the difficulty in predefining the low limit is the range of the economizer performance within each climate zone. A standard designer's rule-of-thumb has been to specify the low-limit as the balance point temperature, but often not lower than 40°F to prevent freezing of coils.

In the previous chapter, the balance point temperature was introduced and is the OA temperature where no heating or cooling is needed; the heat gains of the building equal the heat losses. Cooling, mechanically-produced or otherwise, is required when the OA temperature increases above the balance point temperature. At that point, when provided, an air-side economizer would activate to provide that cooling until its limit of effectiveness is reached. The equation for estimating the balance point temperature includes the effect of climate, but many other factors are constants. This allows the balance point temperature to be predicted for specific climate zones. To observe the relation of the low limit to the balance point temperature, calculations were thus made for each climate zone. The determined temperatures can be found in Table 6.1.

Table 6.1 Calculated balance point temperature for each climate zone for the base case building

Climate Zone	Location	Balance Point Temperature (°F (°C))
Base Case	Lenexa, KS	29.1 (-1.6)
1A	Miami, FL	29.7 (-1.3)
2A	Houston, TX	30.0 (-1.1)
2B	Phoenix, AZ	29.3 (-1.5)
3A	Atlanta, GA	29.2 (-1.6)
3B	Los Angeles, CA	28.9 (-1.7)
3C	San Francisco, CA	28.8 (-1.8)
4A	Baltimore, MD	29.7 (-1.3)
4B	Albuquerque, NM	29.2 (-1.6)
4C	Seattle, WA	28.9 (-1.7)
5A	Chicago, IL	29.4 (-1.4)
5B	Boulder, CO	28.4 (-2.0)
5C	Vancouver, BC	28.6 (-1.9)
6A	Minneapolis, MN	29.3 (-1.5)
6B	Helena, MT	28.4 (-2.0)
7	Duluth, MN	28.8 (-1.8)
8	Fairbanks, AK	28.3 (-2.1)

Chapter IV examined how the energy consumption varies with the low limit. The remainder of this chapter shows how the balance point temperature relates to the low limit. In addition, the energy consumption for the high-limit selected is shown via point "A" on the following figures. Point "B" is the energy consumption with a 55°F (12.8°C) low limit and thus no free cooling. Point "C" shows the energy consumption with the low limit equal to the balance point temperature. Range "D" represents the savings by using an economizer but with no free cooling. Range "E" is the additional savings by operating the economizer with the free cooling. Range "F" shows the total savings when utilizing the air-side economizer fully from the low-limit equal to the balance point. Figure 6.1 shows these points and ranges for the base case in Lenexa, KS. Table 6.2 provides the numerical values for them. Figures 6.2 through 6.17 and Tables 6.3 through 6.18, for the base case building and each of the different climate zones, present their energy consumption and savings potential.

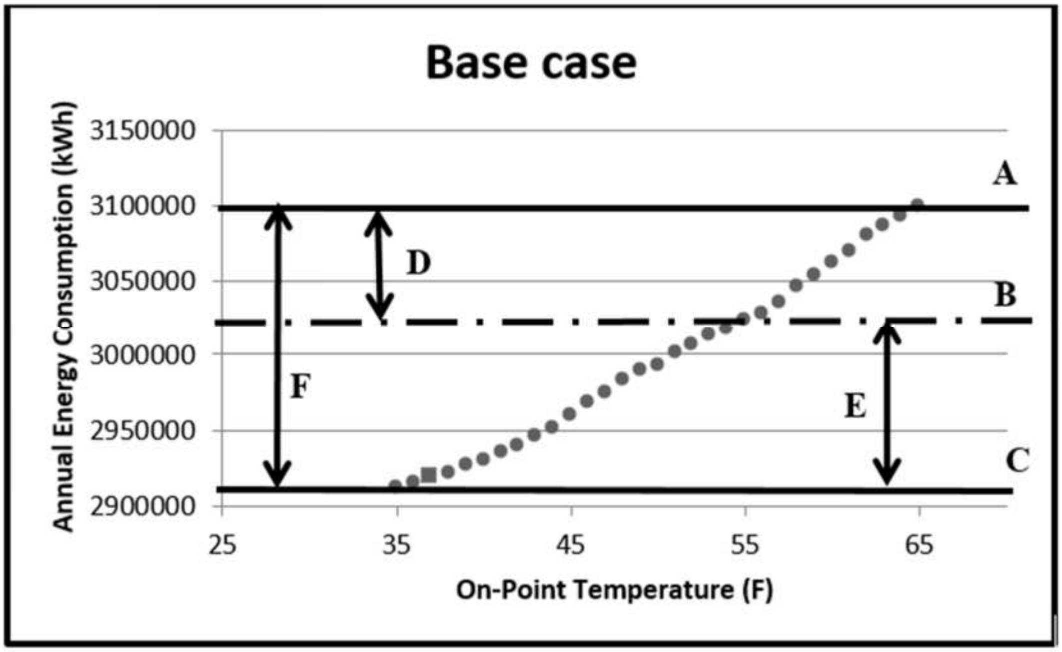

Figure 6.1 Results for the base case in Lenexa, KS with points and ranges of interest

Table 6.2 Base case energy consumption or savings (kWh).

Point or Range	Annual Energy Consumption or Savings
A	3,099,700
B	3,023,500
C	2,894,900
D	76,200
E	128,600
F	204,800

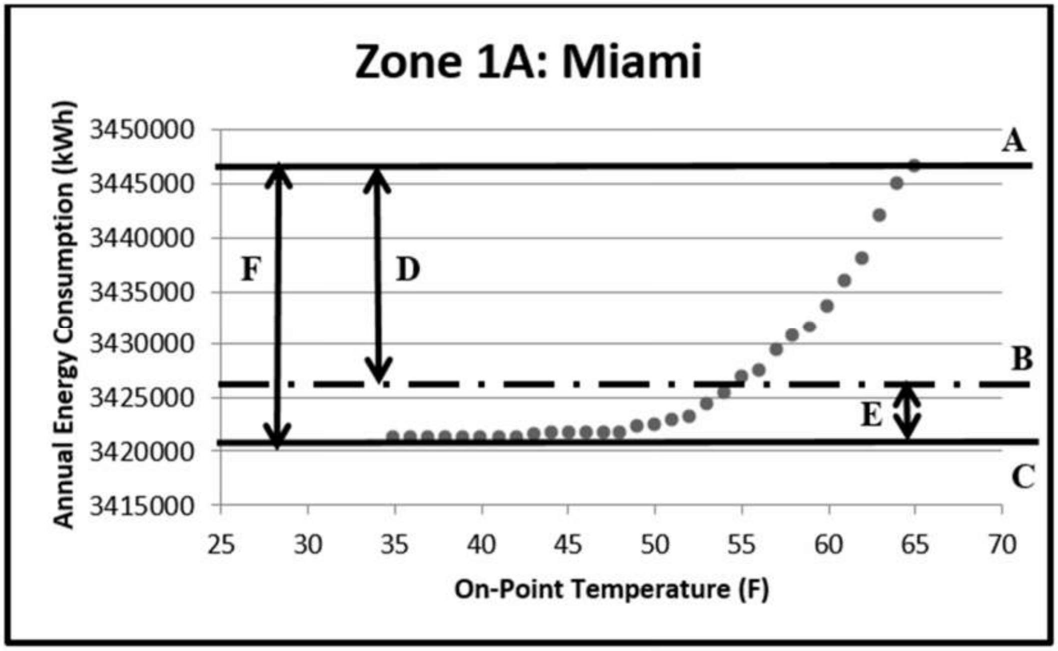

Figure 6.2 Results for Zone 1A in Miami, FL with points and ranges of interest

Table 6.3 Zone 1A energy consumption or savings (kWh).

Point or Range	Annual Energy Consumption or Savings
A	3,446,400
B	3,426,700
C	3,421,200
D	19,700
E	5,500
F	25,200

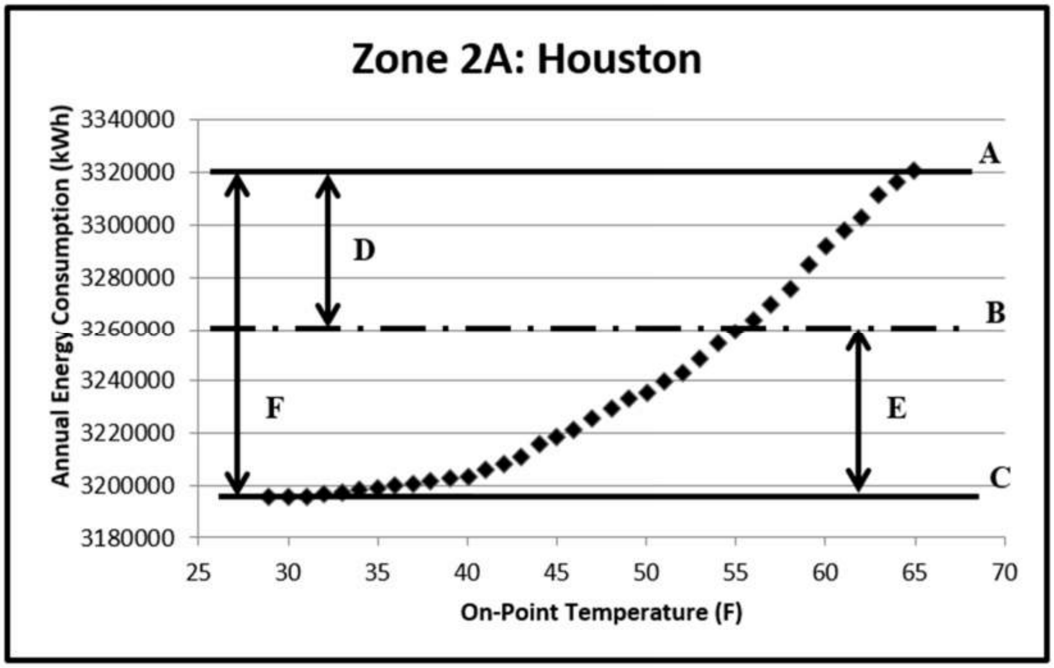

Figure 6.3 Results for Zone 2A in Houston, TX with points and ranges of interest

Table 6.4 Zone 2A energy consumption or savings (kWh).

Point or Range	Annual Energy Consumption or Savings
A	3,320,100
B	3,258,800
C	3,195,700
D	61,300
E	63,100
F	124,400

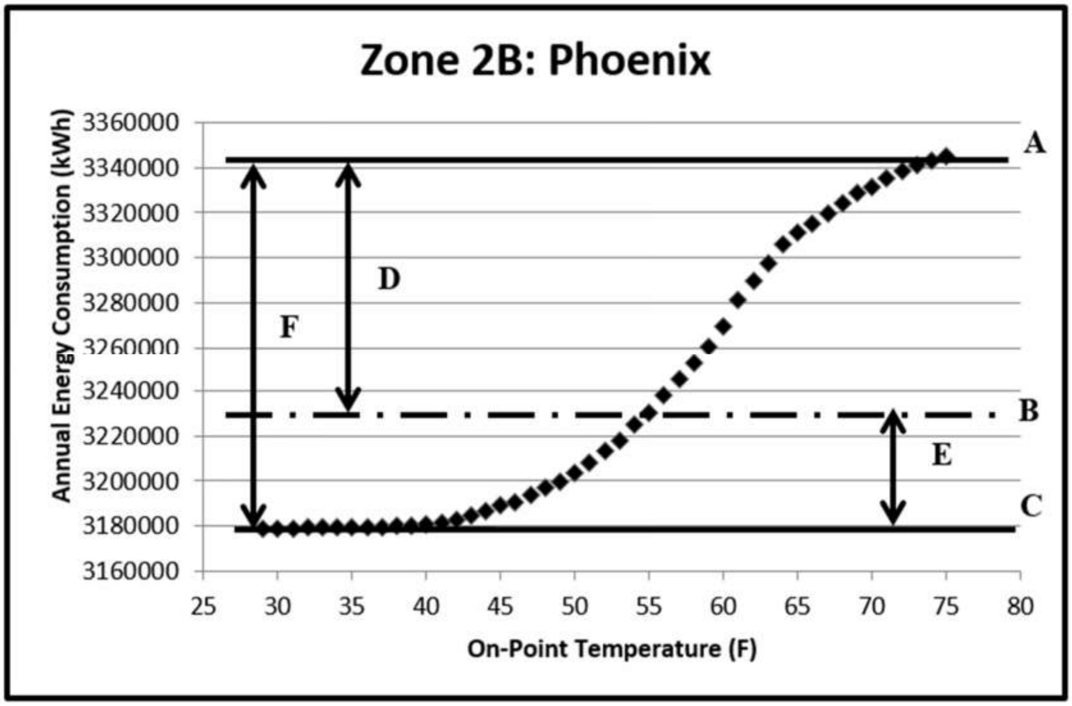

Figure 6.4 Results for Zone 2B in Phoenix, AZ with points and ranges of interest

Table 6.5 Zone 2B energy consumption or savings (kWh).

Point or Range	Annual Energy Consumption or Savings
A	3,344,700
B	3,230,300
C	3,179,200
D	114,400
E	51,100
F	165,500

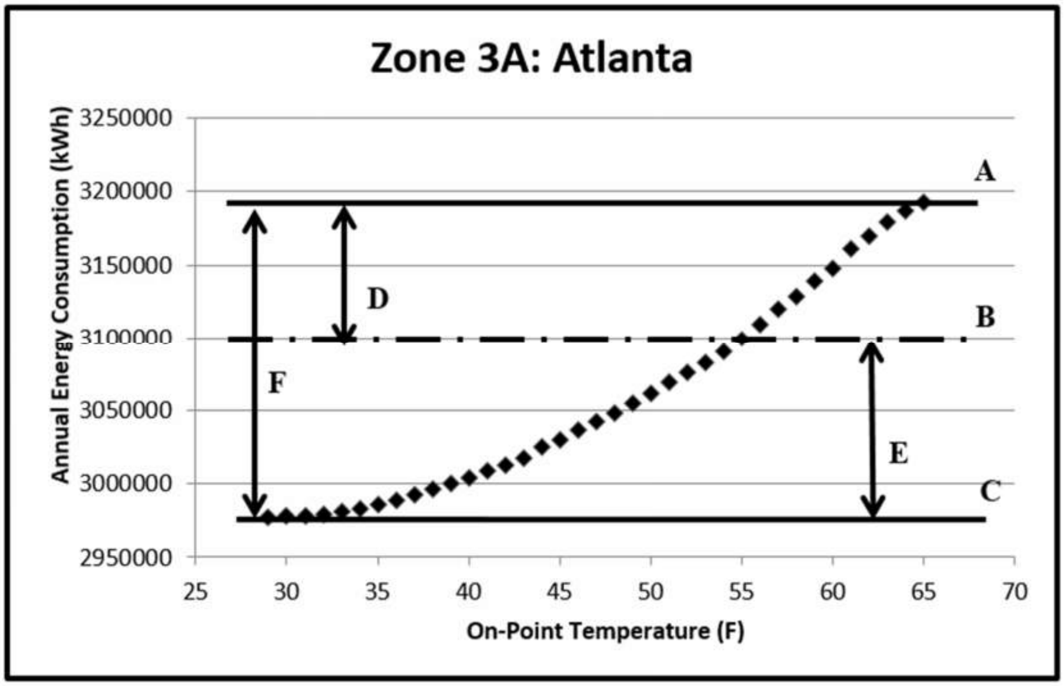

Figure 6.5 Results for Zone 3A in Atlanta, GA with points and ranges of interest

Table 6.6 Zone 3A energy consumption or savings (kWh).

Point or Range	Annual Energy Consumption or Savings
A	3,192,400
B	3,099,200
C	2,977,000
D	93,200
E	122,200
F	215,400

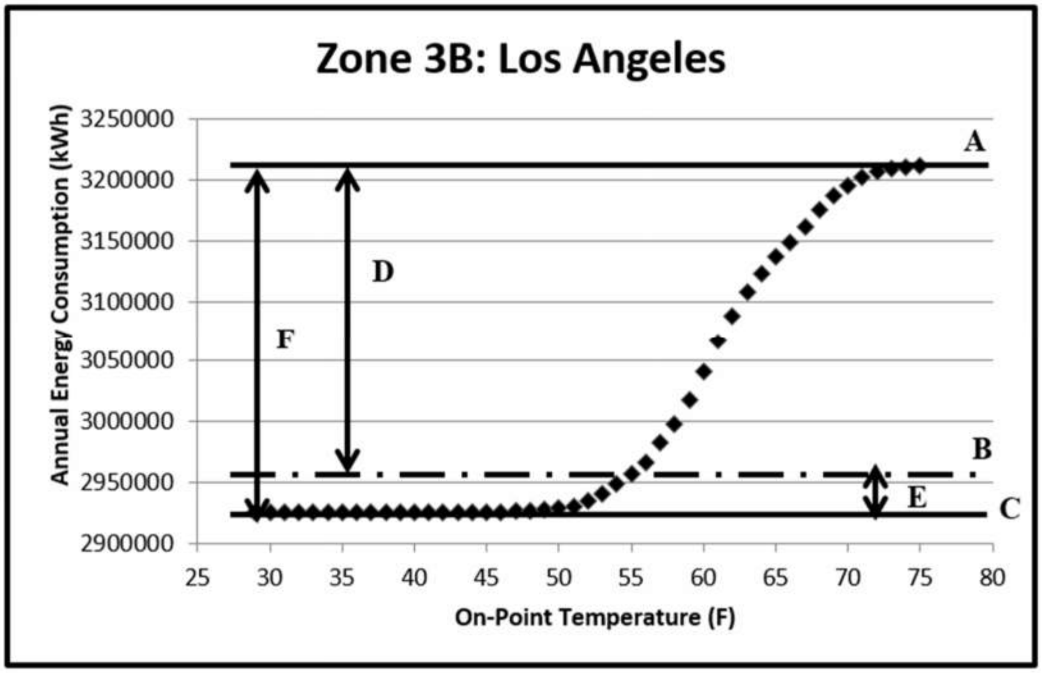

Figure 6.6 Results for Zone 3B in Los Angeles, CA with points and ranges of interest

Table 6.7 Zone 3B energy consumption or savings (kWh).

Point or Range	Annual Energy Consumption or Savings
A	3,211,200
B	2,956,000
C	2,925,000
D	255,200
E	31,000
F	286,200

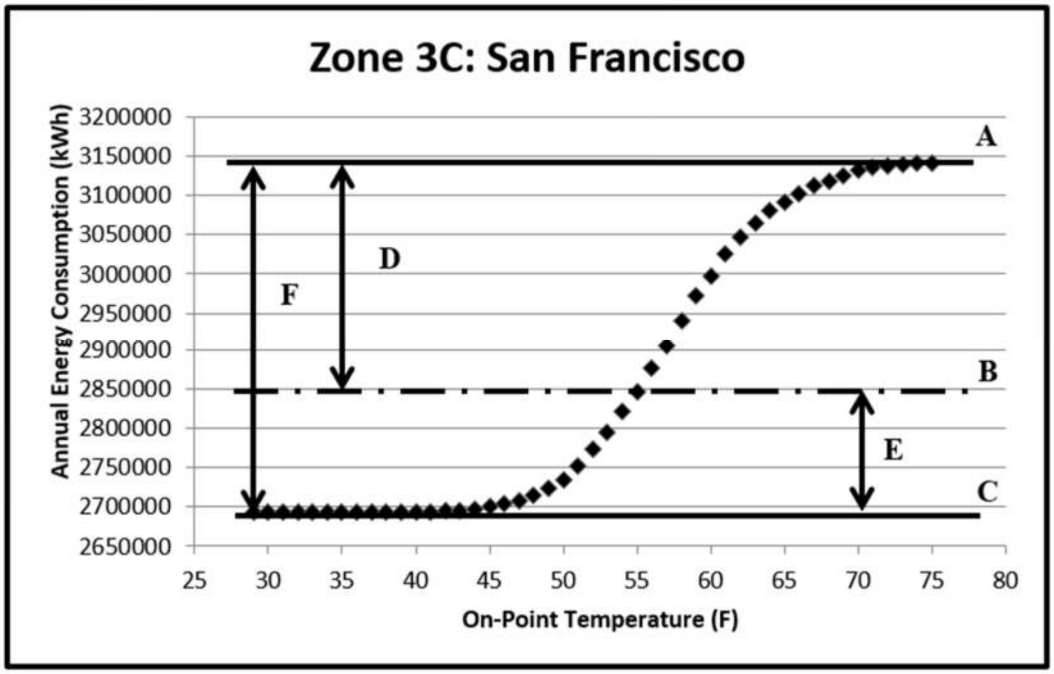

Figure 6.7 Results for Zone 3C in San Francisco, CA with points and ranges of interest

Table 6.8 Zone 3C energy consumption or savings (kWh).

Point or Range	Annual Energy Consumption or Savings
A	3,141,000
B	2,846,500
C	2,693,400
D	294,500
E	153,100
F	447,600

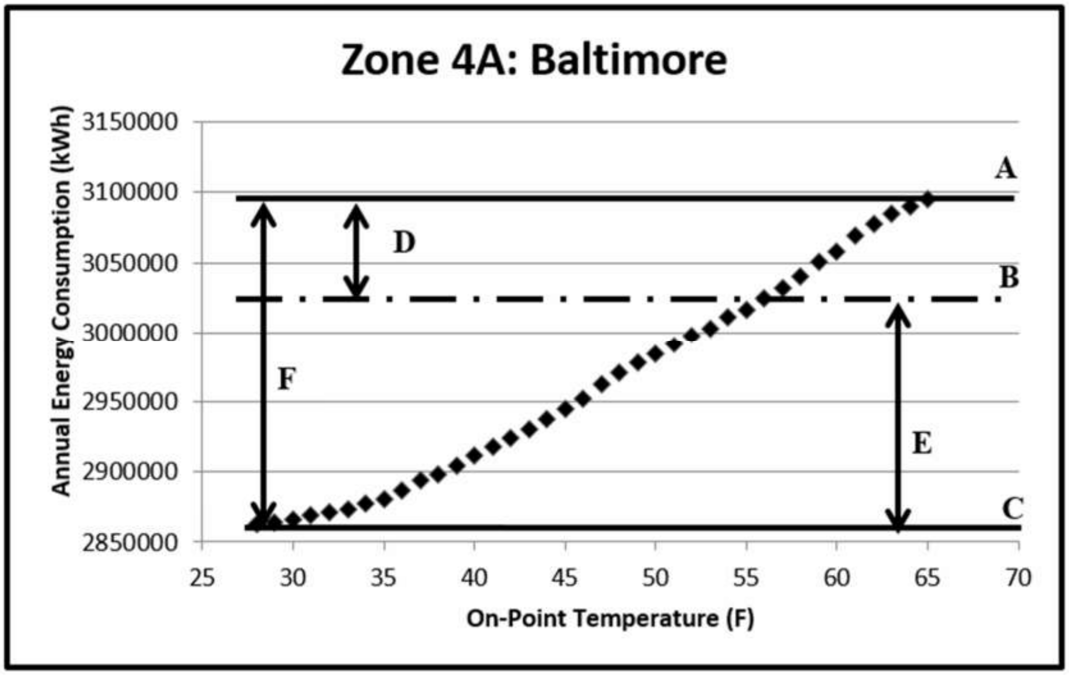

Figure 6.8 Results for Zone 4A in Baltimore, MD with points and ranges of interest

Table 6.9 Zone 4A energy consumption and savings (kWh).

Point or Range	Annual Energy Consumption or Savings
A	3,095,100
B	3,016,800
C	2,869,500
D	78,300
E	150,900
F	229,200

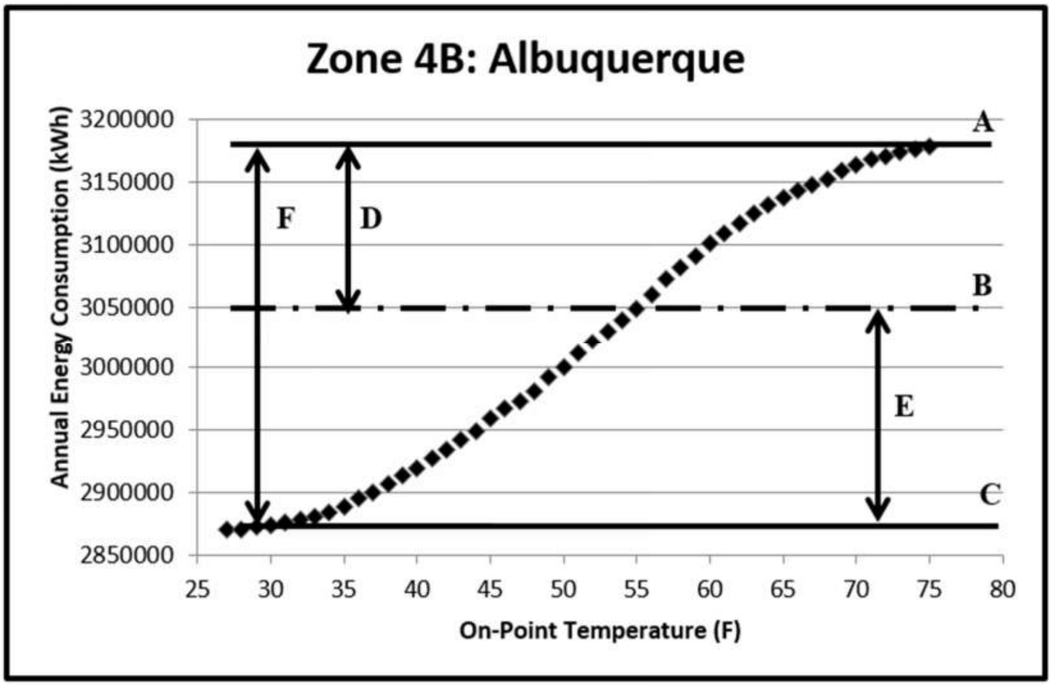

Figure 6.9 Results for Zone 4B in Albuquerque, NM with points and ranges of interest

Table 6.10 Zone 4B energy consumption or savings (kWh).

Point or Range	Annual Energy Consumption or Savings
A	3,178,300
B	3,049,200
C	2,872,600
D	129,100
E	176,600
F	305,700

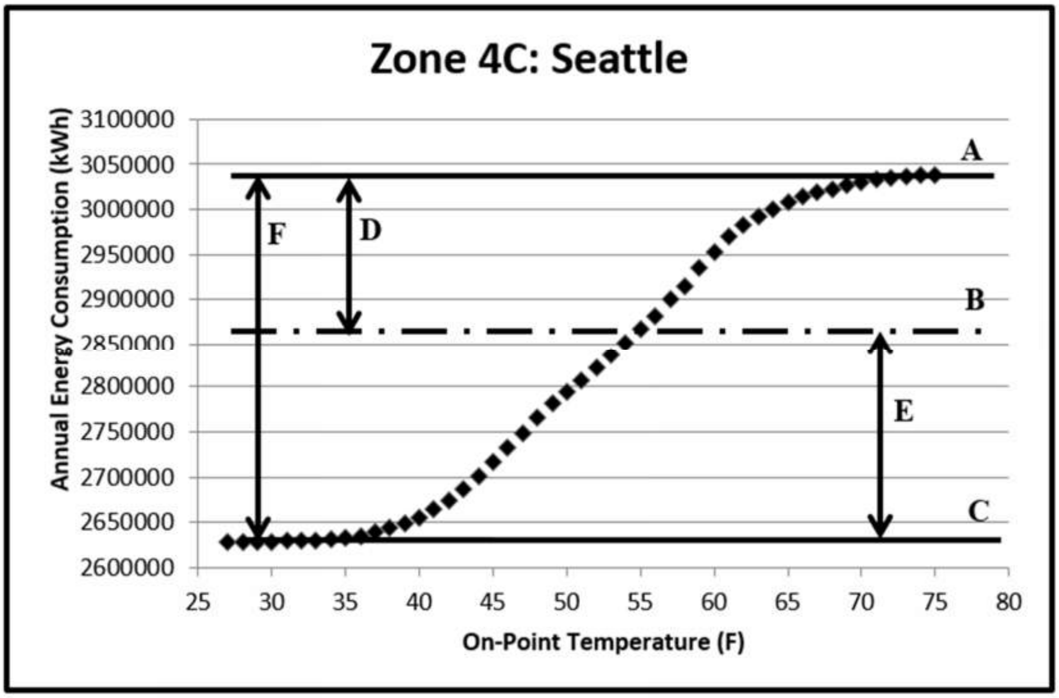

Figure 6.10 Results for Zone 4C in Seattle, WA with points and ranges of interest

Table 6.11 Zone 4C energy consumption or savings (kWh).

Point or Range	Annual Energy Consumption or Savings
A	3,038,100
B	2,866,500
C	2,628,700
D	171,600
E	237,800
F	409,400

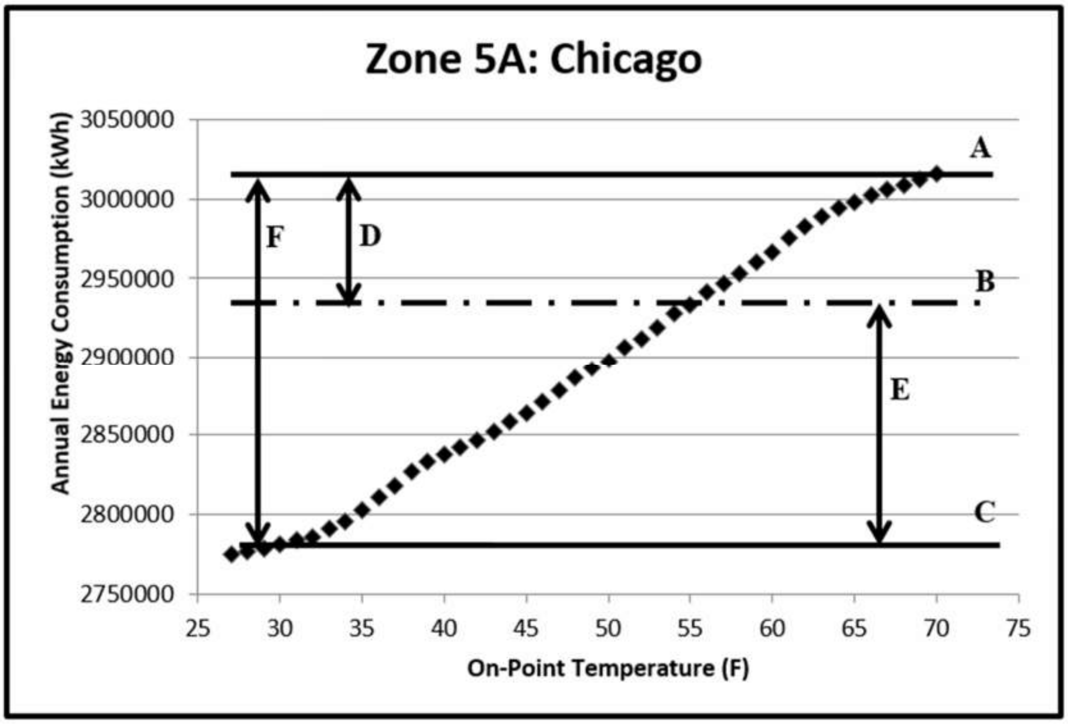

Figure 6.11 Results for Zone 5A in Chicago, IL with points and ranges of interest

Table 6.12 Zone 5A energy consumption or savings (kWh).

Point or Range	Annual Energy Consumption or Savings
A	3,015,600
B	2,933,600
C	2,778,800
D	82,000
E	154,800
F	236,800

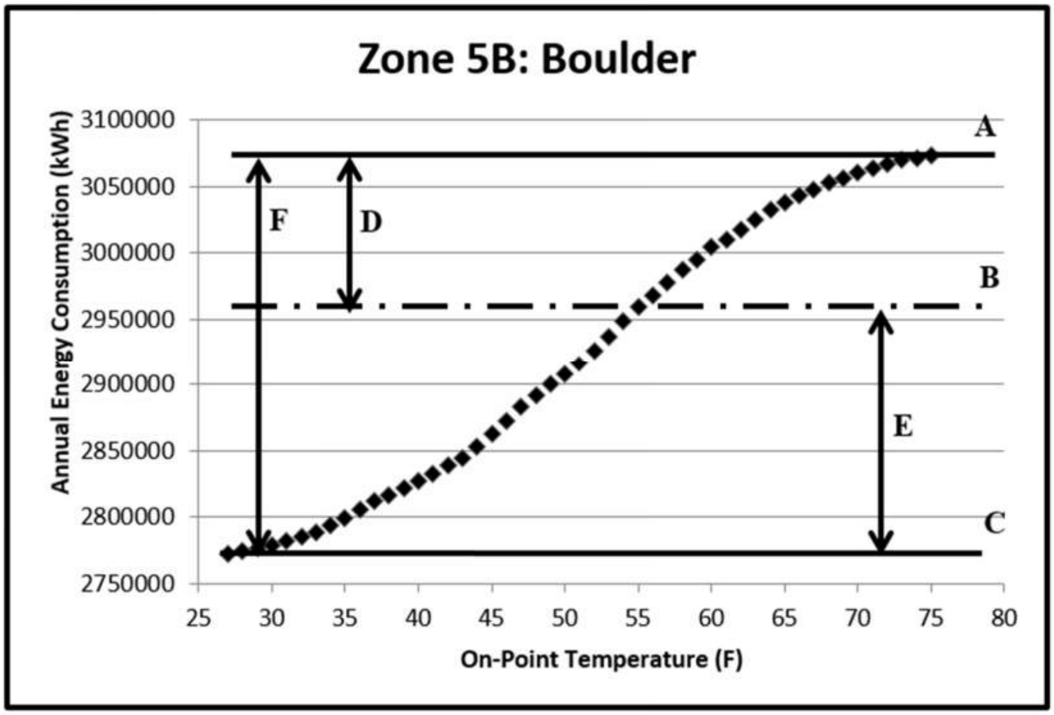

Figure 6.12 Results for Zone 5B in Boulder, CO with points and ranges of interest

Table 6.13 Zone 5B energy consumption or savings (kWh).

Point or Range	Annual Energy Consumption or Savings
A	3,072,800
B	2,959,500
C	2,775,000
D	113,300
E	184,500
F	297,800

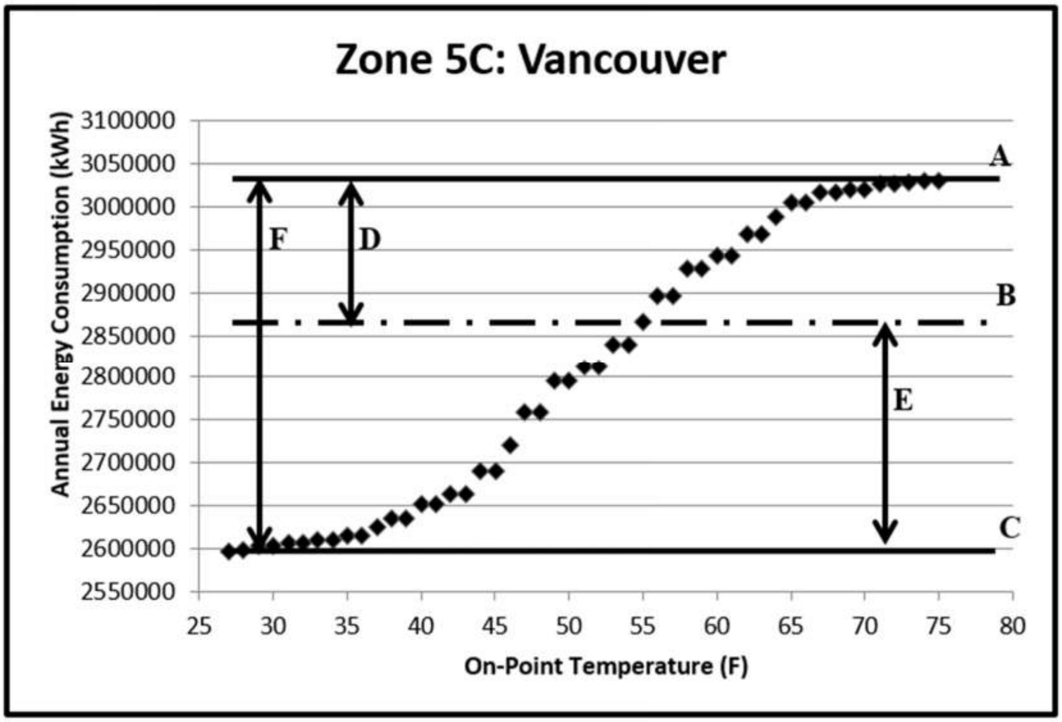

Figure 6.13 Results for Zone 5C in Vancouver, BC with points and ranges of interest

Table 6.14 Zone 5C energy consumption or savings (kWh).

Point or Range	Annual Energy Consumption or Savings
A	3,029,500
B	2,865,700
C	2,602,300
D	163,800
E	263,400
F	427,200

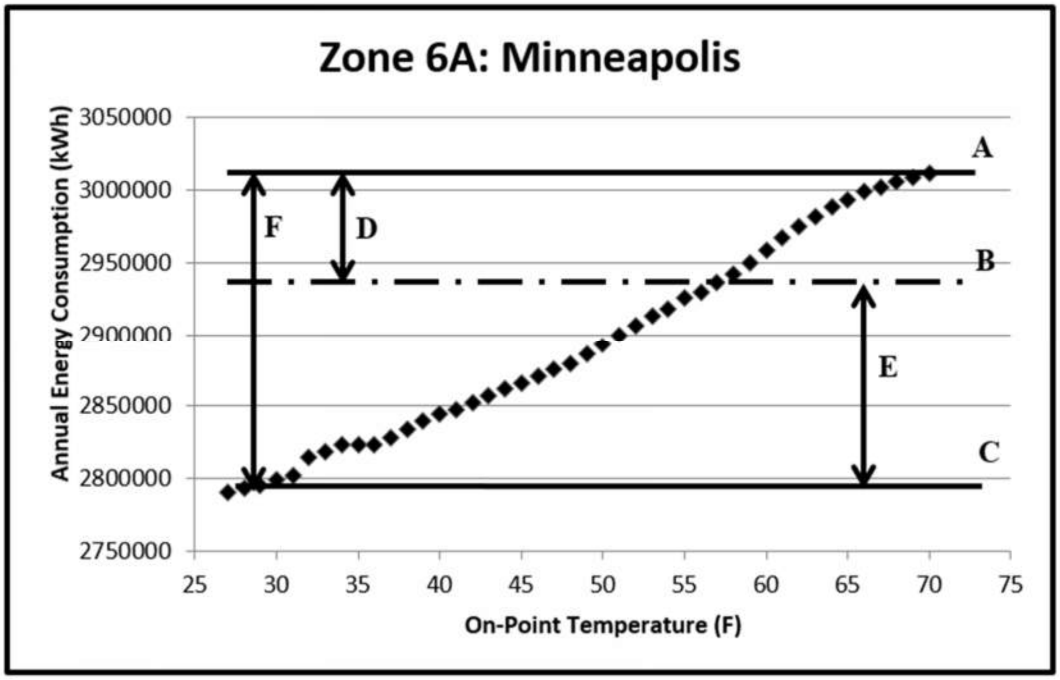

Figure 6.14 Results for Zone 6A in Minneapolis, MN with points and ranges of interest

Table 6.15 Zone 6A energy consumption or savings (kWh).

Point or Range	Annual Energy Consumption or Savings
A	3,071,700
B	2,925,600
C	2,795,500
D	146,100
E	130,100
F	276,200

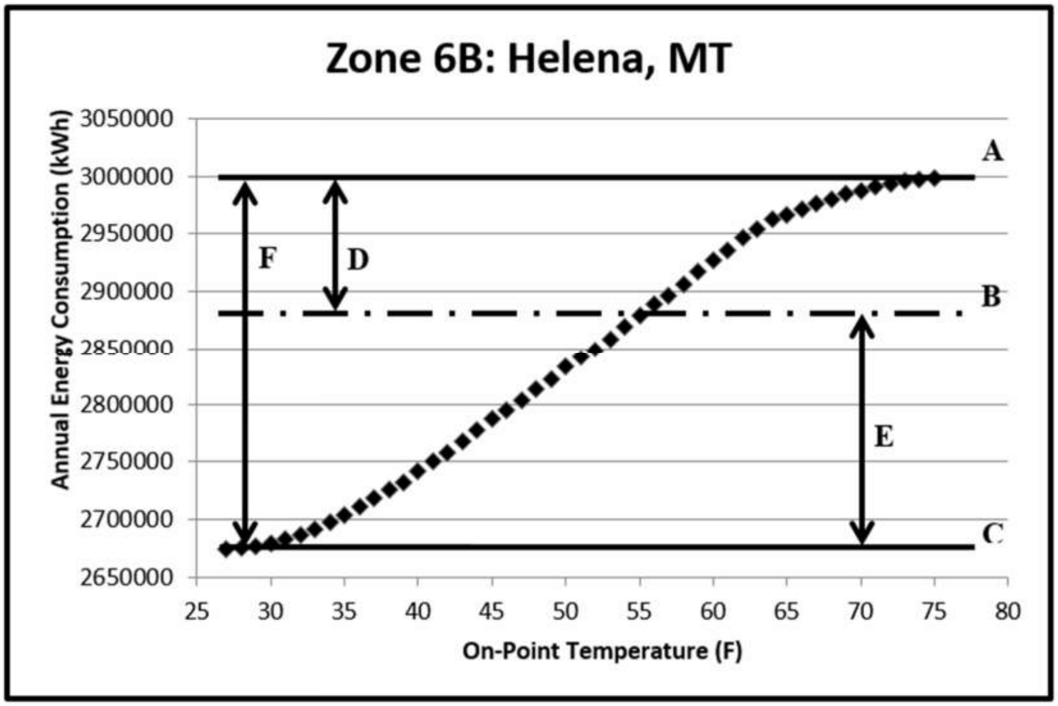

Figure 6.15 Results for Zone 6B in Helena, MT with points and ranges of interest

Table 6.16 Zone 6B energy consumption or savings (kWh).

Point or Range	Annual Energy Consumption or Savings
A	2,998,000
B	2,879,300
C	2,675,200
D	118,700
E	204,100
F	322,800

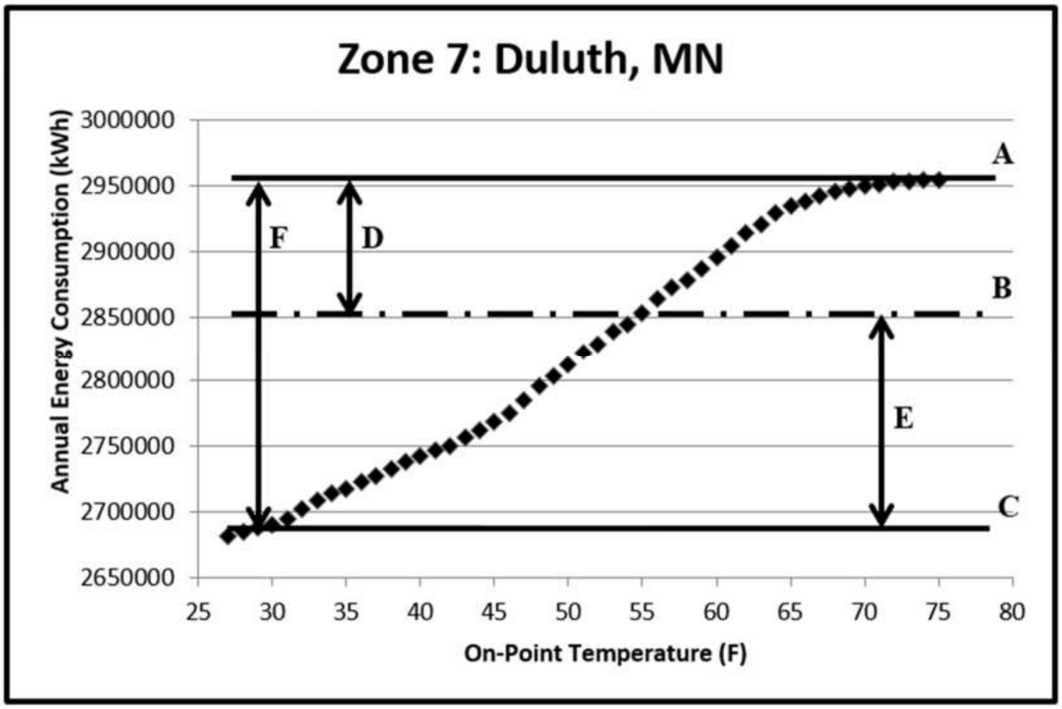

Figure 6.16 Results for Zone 7 in Duluth, MN with points and ranges of interest

Table 6.17 Zone 7 energy consumption or savings (kWh).

Point or Range	Annual Energy Consumption or Savings
A	2,954,300
B	2,853,600
C	2,688,000
D	100,700
E	165,600
F	266,300

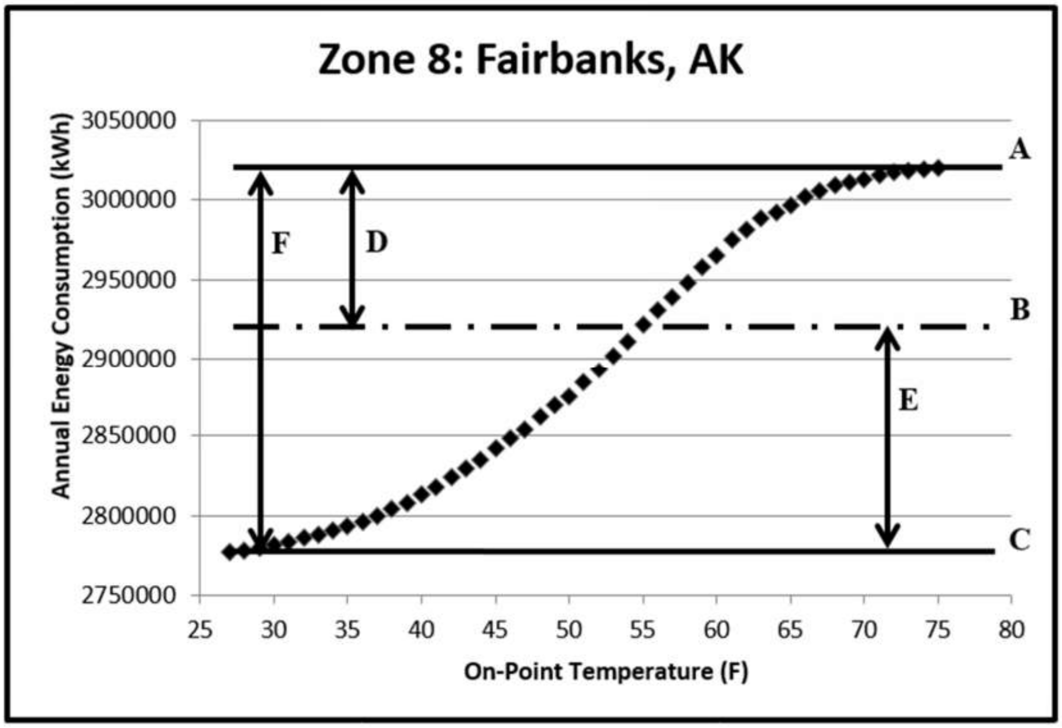

Figure 6.17 Results for Zone 8 in Fairbanks, AK with points and ranges of interest

Table 6.18 Zone 8 energy consumption or savings (kWh).

Point or Range	Annual Energy Consumption or Savings
A	3,019,600
B	2,921,500
C	2,778,200
D	98,100
E	143,300
F	241,400

This chapter's figures and tables indicate that in almost all cases there is a clear low-limit. The calculated balance point temperature for each location corresponded to where the slope of the energy consumption variation became near zero. These results verify the rule-of-thumb used for decades that the low-limit for an air-side economizer should be the balance point temperature of the building.

The balance point temperatures across the climate zones were similar for this building, which was an interesting finding. From Chapter V the largest variable by climate zone was the instantaneous solar load. However, the annual total solar load didn't vary enough across the climate zones to have a large effect on the balance point temperature which is an annual-average value.

Based on the aforementioned cases, a reason that most air-side economizers are not functioning at their highest potential is becoming clear. Designers are specifying or installers are setting too high low-limit setpoints; significant savings can be had by using much lower low-limits that are equal to the balance point temperatures of the buildings, but freeze protection must be included.

CHAPTER VII

CONCLUSIONS AND RECOMMENDATIONS

The purpose of this research study was to examine drybulb air-side economizer low-limits' effect on energy consumption for an office building in many North American climates. The base case selected was a real four-story commercial office building, of typical modern design and construction, in Lenexa, KS. This study used two different modeling programs for the HVAC load and energy simulations -- Trace^{TM}700 and eQUEST. As similar inputs as possible were used with each program, and, for the base case, the results were compared to several years' worth of actual utility data to calibrate the model. Both programs were then used to perform simulations in 16 typical climates of North America. eQUEST, being a more robust program, was used to perform the many simulations needed to evaluate the annual energy consumption when the air-side economizer was activated and its low-limit temperature varied for each of the weather sites. The high-limit temperature was held constant. The simulations' results were then plotted for each of the climate zones. Many of the resulting curves of annual cooling energy use, for each climate zone, had an obvious outside air temperature setting for the low-limit where the reduction in energy consumption became close to zero. The next step in this study was to find a way to calculate this optimal low-limit temperature so that HVAC designers can specify the value. The historical rule-of-thumb is that the low-limit is the balance point temperature. However, the annual-average balance point temperature is not easily calculated -- this was observed to be due to the transient solar heat gain component. The ultimate goal of this project was to create a practical way to perform the calculation. Using the results from a Trace^{TM}700 loads' prediction allowed that to be possible, and performing these load calculations are already routinely done as part of HVAC designs. A Building Balance Point Temperature worksheet was created in this project. The worksheet allows the engineer to enter data directly from a Trace™700 "System Checksums" report or other similar transient HVAC load calculation programs. When utilizing these loads calculations programs it's important for the designer to select an algorithm that uses transient hour-by-hour calculations, with appropriate weather data, so that the buildings' solar and other thermal characteristics are adequately modeled to capture the annual, rather than steady-state "worst case" performance.

In this study, after calculating the balance point temperatures for each of the various climate zones, these values were compared to the curves of annual energy consumption. This comparison verified that the optimal low-limits were consistent with the balance point temperatures calculated. Comparing the results confirmed the decade's old rule-of-thumb to be valid, at least for the building and climates studied. This study resulted in a practical balance point temperature worksheet that allows an HVAC designer to calculate that optimal economizer low-limit. Furthermore, it was found that there is a high level of additional energy and cost savings associated with defining this low-limit lower than a commonly-assumed 40°F (4.4°C) or so. This 40°F assumption likely is from a desire to prevent freezing of water-filled coils in the air handling systems.

Recommendations

Currently, ASHRAE Standard 90.1-2013 gives requirements for airside economizers but with no mention of the low-limit. This study shows that ASHRAE Standard 90.1-2013 needs to include the optimal low-limit. The requirement for the dry-bulb airside economizers' sensors accuracy needs revision too. Currently ASHRAE Standard 90.1-2013 states "Outdoor air, return air, mixed air, and supply air sensors shall be calibrated within the following accuracies: a. Dry-bulb and wet-bulb temperatures shall be accurate to ±2°F over the range of 40°F to 80°F…" (ASHRAE 90.1-2013). This study indicates that optimal low-limits can be 30°F or lower, so the sensors' accuracy should be assured at those reduced temperatures too. A study of the high-limit requirements already in place in ASHRAE Standard 90.1-2013 is also recommended.

If airside economizers' performance were optimized, the energy savings would be vast; expanding the use of the airside economizers to all appropriate buildings is needed too. With buildings now consuming 40% of the primary energy in the U.S., reducing it through expanded use of optimized air-side economizers can noticeably improve energy security and decrease greenhouse gas emissions.

APPENDIX

TRANE TRACE DATA

Entered Values

TRACE® 700 version 6.3

Project Name: Thesis
Dataset Name: C:\Users\Samantha.Pekarscik\Documents\TRACE 700 Projects\THESIS.TRC
Location: 9701 Renner Boulevard
Building Owner: Kiewit
Program User: Samantha Pekarscik
Company: University of Kansas
Comments:

Cooling Design Period: January thru December
Peak Hour Override: 0
Daylight Savings Period:
Summer Period:

Cooling Methodology: TETD-TA1
Heating Methodology: UATD
Infiltration Methodology: Vary with wind speed
Outside Film Methodology: Vary with wind speed
Terrain Methodology: Center of a large city

Room Circ Rate: Medium
Wall Load To Plenum: YES
Building Orientation: 0 degrees from north

Simulation Hours: Reduced year
Calendar Code: Standard (1978)
Energy Simulation Period: January thru December

Location: Topeka, Kansas
Summer Design Dry Bulb: 96.50 °F
Summer Design Wet Bulb: 73.70 °F
Winter Design Dry Bulb: 4.00 °F

Summer Clearness Number: 1.00
Winter Clearness Number: 1.00

Summer Ground Reflectance: 0.20
Winter Ground Reflectance: 0.20
Carbon Dioxide Level: 400 ppm

Force VAV Min => Nominal Ventilation at Design: No
Allow Energy Recovery/Transfer at Design: No
Retest Design Peaks: Yes
Calculate Building Block Loads: No

Close ventilation dampers during unoccupied hours: Yes

ENTERED VALUES
ROOM BY ROOM

Room Description: 2S-5815 Conference Room | **Zone Description: No Zone** | **System Description: System - 001**

GENERAL INFORMATION

Floor Area: 172 ft² Flr-Flr Height: 10.0 ft
Plenum Height: 2.0 ft Height Above Flr:
Slab Cnstr Type: 8" HW Concrete
Room Mass: Time delay based on actual mass
Ceiling R-Value: 1.786 hr·ft²·°F/Btu
Is There Carpet?: YES
Design Clg DB / Drift Point: 75.0 °F / 81.0 °F
Design Htg DB / Drift Point: 70.0 °F / 64.0 °F
Design Relative Humidity: 50 %
Moisture Capacitance: Medium
Clg Tstat: None
Htg Tstat: None
Thermostat Location: Zone Floor Multiplier: 1
Humidistat Location: Room Room Multiplier: 1
CO2 Sensor Location: None
Room Type: Conditioned

PEOPLE

People Type: General Office Space
# of People: 143 sq ft/person
People Sensible: 250 Btu/h
People Latent : 200 Btu/h
People Schedule: Cooling Only (Design)

Workstation: 1.0 workstation/person

LIGHTS

Lighting Type: Recessed fluorescent, not vented, 80% load to space
Fixture Type: RECFL-NV
% Load to RA: 20 %
Lighting Schedule: Cooling Only (Design)
Lighting Amount: 1.1 W/sq ft
Ballast Factor: 1.0

AIRFLOW INFORMATION

	Cooling (Peop-based)	Heating (Area-based)
Vent Type:	Conference/ meeting	Conference/ meeting
Vent Value:	7.50 cfm/person	0.06 cfm/sq ft
Vent Schedule:	Available (100%)	
Infil Type:	Pressurized, Average Const.	Pressurized, Average Const.
Infil Value:	0.30 air changes/hr	0.30 air changes/hr
Infil Schedule:	Available (100%)	
Vav Airflow:		
Vav Sched:	Available (100%)	
Supply:	To be calculated	To be calculated
Aux Supply:	To be calculated	To be calculated
Room Exhaust:		
Rm Exh Sched:	Available (100%)	

Std 62.1-2004

Cooling Ez: Ceiling clg supply, ceiling return 100 %
Heating Ez: Ceiling supply > Trm+15°F(8°C), ceiling return 80 %
Er: Default based on system type

Description	Area/ Amount	Dir	Tilt	Const Type / Schedule	U Value Btu/h·ft²·°F	Alpha	Type / Energy Type	Glass Area ft²	Glass Shade Coef	Glass U Value Btu/h·ft²·°F	External Shading	Internal Shading	Adj Temp/ Grnd Refl	Pct Sen/ Cool Tmp	Pct Rm/ Heat Tmp	Pct Ret/ Perm Len	Rad Frc/ Loss Coef
Misc Load 1	0.500 W/sq ft			Cooling Only (Design)			None							100	100	0	60.00

Room Description: 2S-5819 Office | **Zone Description: No Zone** | **System Description: System - 001**

GENERAL INFORMATION

Floor Area: 117 ft² Flr-Flr Height: 10.0 ft
Plenum Height: 2.0 ft Height Above Flr:
Slab Cnstr Type: 8" HW Concrete
Room Mass: Time delay based on actual mass
Ceiling R-Value: 1.786 hr·ft²·°F/Btu
Is There Carpet?: YES
Design Clg DB / Drift Point: 75.0 °F / 81.0 °F
Design Htg DB / Drift Point: 70.0 °F / 64.0 °F
Design Relative Humidity: 50 %
Moisture Capacitance: Medium
Clg Tstat: None
Htg Tstat: None
Thermostat Location: Zone Floor Multiplier: 1
Humidistat Location: Room Room Multiplier: 1
CO2 Sensor Location: None
Room Type: Conditioned

PEOPLE

People Type: General Office Space
# of People: 143 sq ft/person
People Sensible: 250 Btu/h
People Latent : 200 Btu/h
People Schedule: Cooling Only (Design)

Workstation: 1.0 workstation/person

LIGHTS

Lighting Type: Recessed fluorescent, not vented, 80% load to space
Fixture Type: RECFL-NV
% Load to RA: 20 %
Lighting Schedule: Cooling Only (Design)
Lighting Amount: 1.1 W/sq ft
Ballast Factor: 1.0

AIRFLOW INFORMATION

	Cooling (Peop-based)	Heating (Area-based)
Vent Type:	Office space	Office space
Vent Value:	7.50 cfm/person	0.06 cfm/sq ft
Vent Schedule:	Available (100%)	
Infil Type:	Pressurized, Average Const.	Pressurized, Average Const.
Infil Value:	0.30 air changes/hr	0.30 air changes/hr
Infil Schedule:	Available (100%)	
Vav Airflow:		
Vav Sched:	Available (100%)	
Supply:	To be calculated	To be calculated
Aux Supply:	To be calculated	To be calculated
Room Exhaust:		
Rm Exh Sched:	Available (100%)	

Std 62.1-2004

Cooling Ez: Ceiling clg supply, ceiling return 100 %
Heating Ez: Ceiling supply > Trm+15°F(8°C), ceiling return 80 %
Er: Default based on system type

Description	Area/ Amount	Dir	Tilt	Const Type / Schedule	U Value Btu/h·ft²·°F	Alpha	Type / Energy Type	Glass Area ft²	Glass Shade Coef	Glass U Value Btu/h·ft²·°F	External Shading	Internal Shading	Adj Temp/ Grnd Refl	Pct Sen/ Cool Tmp	Pct Rm/ Heat Tmp	Pct Ret/ Perm Len	Rad Frc/ Loss Coef
Misc Load 1	0.500 W/sq ft			Cooling Only (Design)			None							100	100	0	60.00

ENTERED VALUES
ROOM BY ROOM

Room Description: 2S-6005 Cubes | **Zone Description: No Zone** | **System Description: System - 001**

GENERAL INFORMATION

Floor Area: 406 ft² FlrFlr Height: 10.0 ft
Plenum Height: 2.0 ft Height Above Flr:
Slab Cnstr Type: 8" HW Concrete
Room Mass: Time delay based on actual mass
Ceiling R-Value: 1.786 hr·ft²·°F/Btu
Is There Carpet?: YES
Design Clg DB / Drift Point: 75.0 °F / 81.0 °F
Design Htg DB / Drift Point: 70.0 °F / 64.0 °F
Design Relative Humidity: 50 %
Moisture Capacitance: Medium
Clg Tstat: None
Htg Tstat: None
Thermostat Location: Zone Floor Multiplier: 1
Humidistat Location: Room Room Multiplier: 1
CO2 Sensor Location: None
Room Type: Conditioned

PEOPLE

People Type: General Office Space
# of People: 143 sq ft/person
People Sensible: 250 Btu/h
People Latent : 200 Btu/h
People Schedule: Cooling Only (Design)
Workstation: 1.0 workstation/person

LIGHTS

Lighting Type: Recessed fluorescent, not vented, 80% load to space
Fixture Type: RECFL-NV
% Load to RA: 20 %
Lighting Schedule: Cooling Only (Design)
Lighting Amount: 1.1 W/sq ft
Ballast Factor: 1.0

AIRFLOW INFORMATION

	Cooling (Peop-based)	Heating (Area-based)
Vent Type:	Office space	Office space
Vent Value:	7.50 cfm/person	0.06 cfm/sq ft
Vent Schedule:	Available (100%)	
Infil Type:	Pressurized, Average Const.	Pressurized, Average Const.
Infil Value:	0.30 air changes/hr	0.30 air changes/hr
Infil Schedule:	Available (100%)	
Vav Airflow:		
Vav Sched:	Available (100%)	
Supply:	To be calculated	To be calculated
Aux Supply:	To be calculated	To be calculated
Room Exhaust:		
Rm Exh Sched:	Available (100%)	

Std 62.1-2004

Cooling Ez: Ceiling clg supply, ceiling return 100 %
Heating Ez: Ceiling supply > Trm+15°F(8°C), ceiling return 80 %
Er: Default based on system type

Description	Area/ Amount	Dir	Tilt	Const Type / Schedule	U Value Btu/h·ft²·°F	Alpha	Type / Energy Type	Glass Area ft²	Shade Coef	U Value Btu/h·ft²·°F	External Shading	Internal Shading	Adj Temp/ Grnd Refl	Pct Sen/ Cool Tmp	Pct Rm/ Heat Tmp	Pct Ret/ Perm Len	Rad Frc/ Loss Coef
Wall - 1	230 ft²	270	0	12" Conc, 5" Ins (int)	0.0525	0.90											
Opening - 1				Window			6mm Dbl Ref C Tint-M	126	0.24	0.51	Overhang - None	None	0.00				
Misc Load 1	0.500 W/sq ft			Cooling Only (Design)			Electricity							100	100	0	60.00

Room Description: 2S-6008 Office | **Zone Description: No Zone** | **System Description: System - 001**

GENERAL INFORMATION

Floor Area: 112 ft² FlrFlr Height: 10.0 ft
Plenum Height: 2.0 ft Height Above Flr:
Slab Cnstr Type: 8" HW Concrete
Room Mass: Time delay based on actual mass
Ceiling R-Value: 1.786 hr·ft²·°F/Btu
Is There Carpet?: YES
Design Clg DB / Drift Point: 75.0 °F / 81.0 °F
Design Htg DB / Drift Point: 70.0 °F / 64.0 °F
Design Relative Humidity: 50 %
Moisture Capacitance: Medium
Clg Tstat: None
Htg Tstat: None
Thermostat Location: Zone Floor Multiplier: 1
Humidistat Location: Room Room Multiplier: 1
CO2 Sensor Location: None
Room Type: Conditioned

PEOPLE

People Type: General Office Space
# of People: 143 sq ft/person
People Sensible: 250 Btu/h
People Latent : 200 Btu/h
People Schedule: Cooling Only (Design)
Workstation: 1.0 workstation/person

LIGHTS

Lighting Type: Recessed fluorescent, not vented, 80% load to space
Fixture Type: RECFL-NV
% Load to RA: 20 %
Lighting Schedule: Cooling Only (Design)
Lighting Amount: 1.1 W/sq ft
Ballast Factor: 1.0

AIRFLOW INFORMATION

	Cooling (Peop-based)	Heating (Area-based)
Vent Type:	Office space	Office space
Vent Value:	7.50 cfm/person	0.06 cfm/sq ft
Vent Schedule:	Available (100%)	
Infil Type:	Pressurized, Average Const.	Pressurized, Average Const.
Infil Value:	0.30 air changes/hr	0.30 air changes/hr
Infil Schedule:	Available (100%)	
Vav Airflow:		
Vav Sched:	Available (100%)	
Supply:	To be calculated	To be calculated
Aux Supply:	To be calculated	To be calculated
Room Exhaust:		
Rm Exh Sched:	Available (100%)	

Std 62.1-2004

Cooling Ez: Ceiling clg supply, ceiling return 100 %
Heating Ez: Ceiling supply > Trm+15°F(8°C), ceiling return 80 %
Er: Default based on system type

Description	Area/ Amount	Dir	Tilt	Const Type / Schedule	U Value Btu/h·ft²·°F	Alpha	Type / Energy Type	Glass Area ft²	Shade Coef	U Value Btu/h·ft²·°F	External Shading	Internal Shading	Adj Temp/ Grnd Refl	Pct Sen/ Cool Tmp	Pct Rm/ Heat Tmp	Pct Ret/ Perm Len	Rad Frc/ Loss Coef
Misc Load 1	0.500 W/sq ft			Cooling Only (Design)			None							100	100	0	60.00

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